SEP 7

Heating · Piping and Air Conditioning

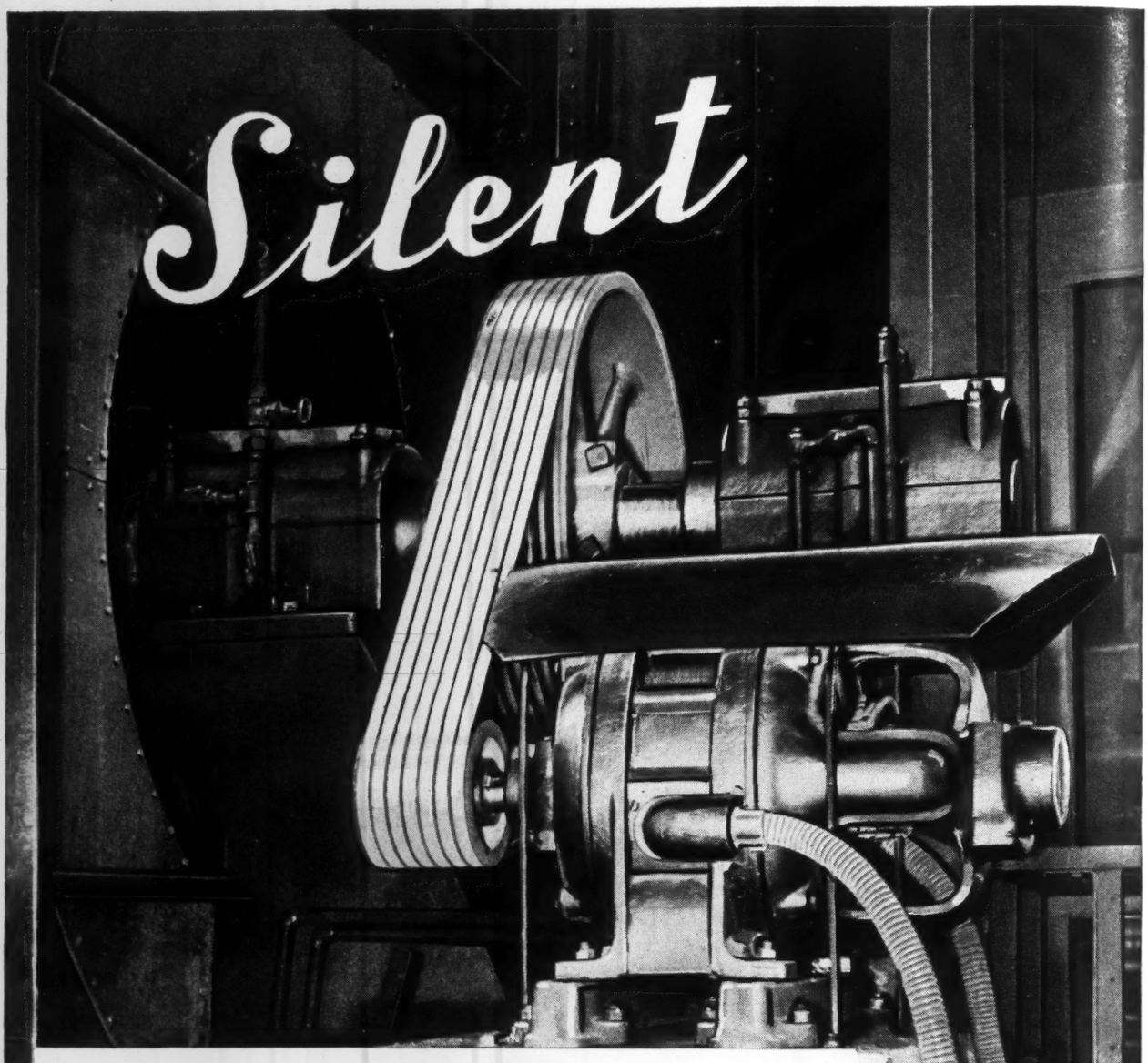
Effective, Economical Plant Heating
Promoted by Modern Unit Heaters

Status of Piping Standardization

Two-Year-Old Building Partly Air Conditioned

Include in the Public Works Program Modernization of Mechanical Services

Journal Section of the A. S. H. V. E.



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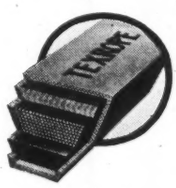

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Heating · Piping and Air Conditioning

THE EDITOR'S PAGE

DURING the past decade or so, unit heaters have firmly established themselves as a means of effective plant heating with economy. In the article starting on page 445 this month are assembled experiences of several plants with unit heaters. In addition, two check lists appear, one designed to be of aid when the suitability of unit heaters to a plant not now equipped with them is being considered. The other check questions are aimed to assist the modernization of an existing unit-heater system. Future articles on this subject will present similar information.

A PLAN to include in the $3,300,000,000 government public-works program the modernization of mechanical services in existing properties is published in this issue with an analysis of its fundamental soundness, its adaptability to aiding the government in accomplishing the objects of the program.

In addition, proof of the plan's practicability is given by means of case records from the files of consulting engineers. These examples—each referring to a specific building—show how an investment in well-planned modernization of a heating, piping, or air conditioning system is self-liquidating; how it will reduce (rather than increase) future expense; how it will provide employment not only for the manual laborer but for engineers as well.

Many more such examples could be mentioned almost off-hand by any consulting engineer. A nation-wide survey of existing properties as proposed in the plan would bring to light countless opportunities for self-liquidating, cost-reducing modernization work.

IF you wanted to get the complete story of national standardization of piping, Sabin Crocker would be one of the best men in the country to go to; he has been active in piping standardization work for ten years, is a member of several of the committees which are working on standardization projects.

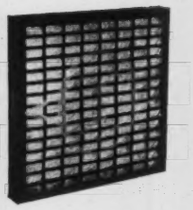
That's just what we have done. Some four years ago, Mr. Crocker wrote an article on this subject for HEATING, PIPING AND AIR CONDITIONING. Many new standards have been added since then, others are now in process. In his series of articles starting this month, Mr. Crocker gives a complete picture of piping standardization today, takes you behind the scenes with his explanations of the reasons back of the standards.

COMPARABLE to the authority with which Mr. Crocker speaks on piping standardization is the standing of R. H. Heilman when it comes to the heat losses from piping, and insulation. To this issue, therefore, Mr. Heilman contributes a paper on heat transmission from copper pipe and tubing, about which some confusion exists. This straight-forward analysis is clear, concise; the author's research work (some of it made especially for this article) is drawn upon extensively.

THE modern office building is equipped with air conditioning; air conditioning offers so many advantages to both the building management and the tenant that the building without it can no longer be said to be up-to-date, thoroughly modern.

Fortunately, existing buildings as well as new can be air conditioned; often the requirements can be met by air conditioning only several floors of an existing structure.

That's the situation in the example described in M. O'Connell's article this month. Several floors of a New York office building were recently equipped with an air-conditioning system. His explanation of the solution of the problems met on this typical job is thorough; you'll find it of interest.



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HEATING, PIPING and AIR CONDITIONING

Vol. 5, No. 9 September, 1933

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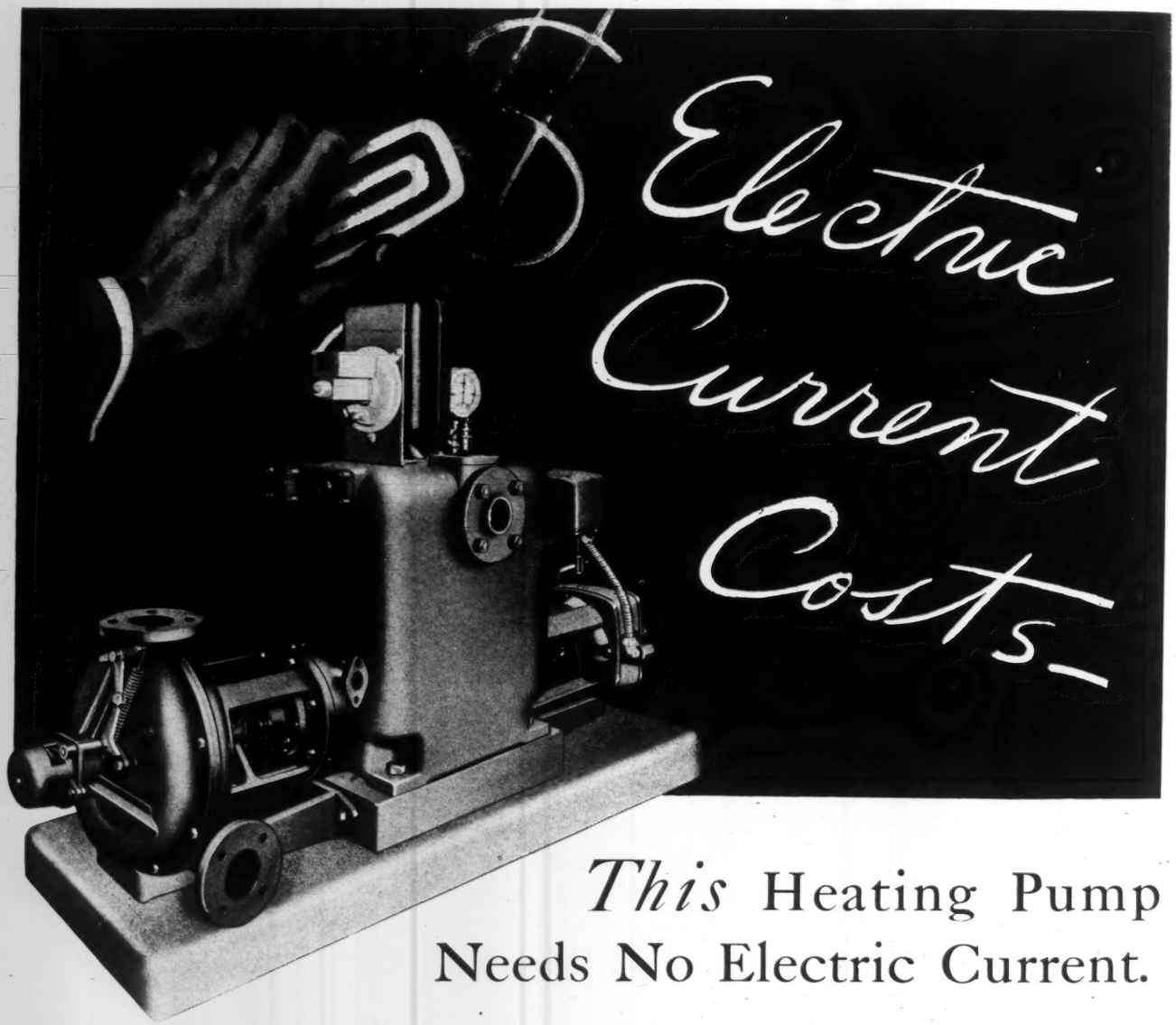

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And remember that this saving in electric current is only one of the economical features offered by the Jennings Vapor Turbine.

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Most important of all, each and every radiator is maintained at top-notch heating efficiency.

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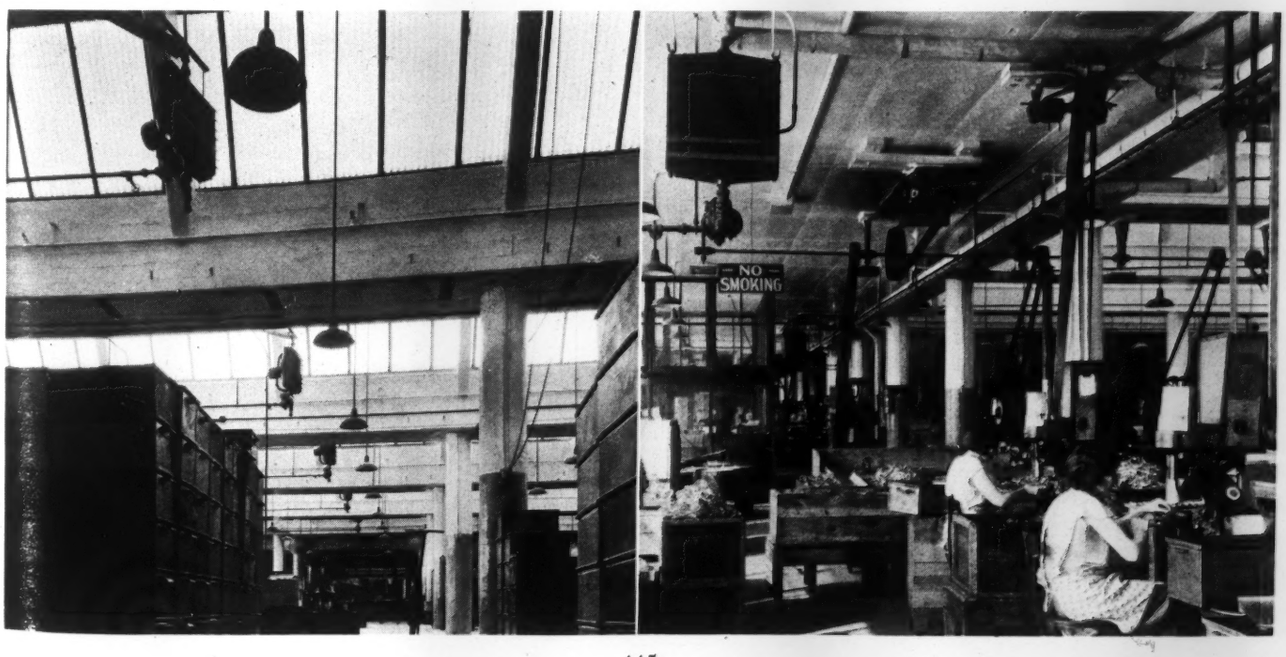
Heating · Piping
and Air Conditioning
September, 1933

Modern Unit Heaters Promote Effective Plant Heating with Economy

Examples Show Opportunities for Saving Money in Industrial Heating . . . Check Lists Aid Application of Unit Heaters

UNIT heaters and their industrial applications are "live" subjects at the present time for a number of reasons. Plant heating systems are now being (or should be) put into final shape for satisfactory, economical operation during the coming heating season. With increased industrial activity and the stress which has been placed on the need for efficient plant operation by NRA, it is probable that the tuning-up process will be more widespread and thorough in plants of almost every industry than has been the case for a long while. Add to this current activity in developing and improving unit heaters themselves, and it is evident that the sub-

Fig. 1—Two views of the unit heaters in the plant of Edward Katzinger Company, a representative example of this modern method of industrial space heating



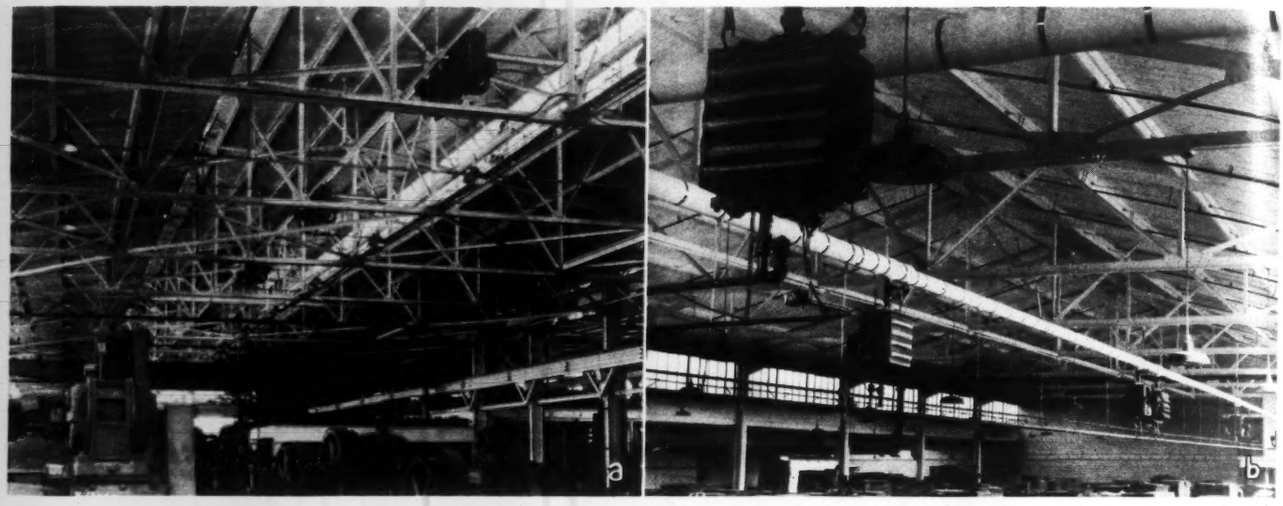


Fig. 2—*a* shows part of the 64 unit heaters installed in the plant of the Kelsey-Hayes Wheel Company at Detroit; the units are 44 ft above floor level. In *b* some of the 33 units in the Woodall Industries, Inc., plant are shown

ject deserves attention from those concerned with the heating of plants and factories.

It is the purpose of this article (and others to follow in future issues) to present needed information on the adaptability of modern unit heaters to solving plant heating problems. A number of cases describing the results which have been obtained in unit-heater equipped plants are reported to indicate what can be done toward promoting efficiency with this type of industrial heating.

Two check-lists also have been prepared and appear

Guide for Determining Suitability of Unit Heaters

[*The following check questions suggest the points to be considered in surveying a plant's heating to see if the installation of unit heaters would be advantageous. If the answers are "yes," suitability of unit heaters to meet the plant heating problem is indicated.*]

Have changed plant conditions presented new requirements which might well be met by unit heaters?.....

(As with any other service, a change in plant lay-out or function may require a change in the duties of the heating system.)

Is temperature distribution now uneven? Are there cold spots in the plant?.....

(An advantage of unit heaters is that the heat may be directed where wanted.)

Are heat losses through the roof proportionately large? Is there wide difference between floor and ceiling temperatures?.....

(Savings of 20 to 50 per cent in heating costs with unit heaters have been reported; these savings are usually greatest where the proportion of overhead heat loss to other heat losses is the greatest. Unit heaters will maintain a relatively warm zone near the floor and a relatively cold zone near the ceiling.)

Would more rapid heating-up be desirable?.....

(Unit heaters offer high heat-transfer capacity in small space with rapid heating and wide distribution of heat, readily controlled.)

Would better control be desirable?.....

(Adaptability to many forms of control is a feature of unit heaters. One authority has stated that 10 to 30 per cent of the fuel is wasted by overheating and open windows when workmen are relied upon to regulate the temperature manually; that the cost of control is saved in less than a year.)

Is economy in investment sought?.....

(The high heat-transfer capacity of a unit-heater system offers opportunities for conserving in the initial cost of a plant heating system.)

Is a saving in the space required for the heating equipment desirable?.....

(Units are available in many types for floor and suspended mounting. They may be located to require no floor space or to fit in with plant requirements to use no space required for production equipment.)

Is simplification of the piping system wanted?.....

(Units offer a chance for simplification of the necessary piping as fewer heat-emitting surfaces are necessary.)

Would ventilation be desirable?.....

(By installing a fresh-air connection, outdoor air can be supplied the unit, heated to the proper temperature and discharged to the working space.)

Is it desired to use high-pressure steam for heating?.....

(Units are available for operation with high-pressure steam.)

Is it desired to use low-pressure steam for heating?.....

(Economy in heating can be had by using low-pressure exhaust steam, for instance, reducing high-pressure steam for make up when necessary.)

Is it desired to use a hot-water system?.....

(Units may operate with hot water.)

Is it desired to use gas?.....

(Gas-fired unit heaters may be had in several types.)

Is it desired to use electric heat?.....

(Electric unit heaters are available. Auxiliary electric heating with low-cost off-peak power has proved a money saver in many plants.)

Maintenance Guide for Plant Unit-Heater Systems

[*These check questions will aid in inspecting a unit-heater system and will suggest what should be done to put a faulty system in shape for efficient operation during the coming heating season. They will also suggest other points that might well be checked in the individual system. If answer to any question is "no," it will pay to apply the proper remedy.*]

Are there enough units properly placed?.....
(Some types of units will project their heating effect 30 to 200 ft, have successfully been placed as far as 400 ft from each other. Other types may be placed 30 to 100 ft apart. Sometimes rotational circulation of the air in the space heated is desired, one unit picking up the air from another. No hard-and-fast rule applies, judgment being required. Limitations of the building, exposed areas, openings, etc., must of course be considered in location of the units.)

Do the units have the proper capacities?.....
(Units must provide for heat loss from the building and for heating any outdoor air from outside to inside temperature. Ratings are usually on basis of 2-lb steam pressure and 60-F* entering air temperature; capacity increases with increased steam pressure, decreases with increasing entering air temperature. Capacity slightly less with superheated steam, negligible up to 50 deg superheat. Consideration must be given to reduced heat and air capacity due resistance from outside air intakes, filters, discharge ducts, etc., if used.)

Is the direction of discharge of each unit o. k.?.....
(Hot blasts of air in working zone are usually undesirable. However, for economy, the heated air should be brought down as far as possible.)

Are units quiet?.....
(Allowable noise depends upon the requirements and use of the building heated. Quietness is a function of several variables besides speed and all should be considered. Support should be such as to prevent vibration.)

Are the motors of the best type and speed for the particular service? Is the drive as a whole efficient and trouble-free? .....
(Bearings in alignment? Lubrication o. k.? Proper voltage to motor? Electrical circuit properly protected?)

Does the control prevent the unit delivering cold air? Is over-heating prevented? .....
(Unit heaters are particularly adapted to automatic control, can be operated economically with the proper control arrangements. A variety of possible control hook-ups permit almost any type of operation to be attained.)

Is the steam pressure (or water temperature) supplied the unit o. k.?.....
(Steam pressures below 5 lb satisfactory for recirculating heaters. When air at temperatures below freezing may be taken in, pressure of not less than 5 lb should be maintained in heating surface. A low-presure unit must not be supplied with high-pressure steam. If a pressure-reducing valve is used, is there a relief valve in low-pressure line between pressure-reducing valve and the unit to relieve pressure if reducing valve sticks?)

Have proper provisions been made for venting air from both units and piping? .....
(Are the air valves large enough? Are there any air pockets?)

Is supply line large enough? Is it pitched back from unit correctly? Does supply valve open full?.....
(Supply piping should be arranged so that condensation from mains, branches, and risers does not pass through heater or collect above the supply valve when closed.)

Are joints in the piping near the unit screwed up tightly?

Have expansion and contraction in both units and piping been adequately provided for?.....
(Such construction problems as tubes pulling away from headers, etc., have been solved in modern unit heaters.)

Is return line of proper size and does it leave unit without being bushed down? .....

Is unit properly drained? Are traps the right size and proper type for the system? Working freely? Cooling legs necessary? If so, are they provided for?.....
(There should be sufficient drop from bottom of unit—12 in.—to float or combination float-and-thermostatic trap to insure sufficient head for removing condensation. Bottom of unit must be far enough—30 in.—above boiler water-line to prevent steam pressure from backing water into unit.)

Are strainer and scale pockets installed?.....
(Strainer in supply connection protects the relatively small and valuable tube area.)

Is heat-transfer surface free of dirt accumulation? Hot over entire surface? .....

**A. S. H. V. E. Standard Code for Testing and Rating Steam Unit Heaters, A. S. H. V. E. Transactions,* Vol. 36, 1930. For constants for determining capacities at various steam pressures and entering air temperatures see pages 404 and 405, *A. S. H. V. E. Guide,* 1933.

on these pages. One of them suggests the points to consider when the application of unit heaters to a plant not now equipped with them is being studied. The other serves a similar purpose when an existing industrial unit-heater system is under consideration for modernizing or maintenance attention. These summarized guides to the use of unit heaters in industry can be utilized to advantage; they insure that nothing that should be considered will be over-looked and suggest the making of a survey which will, in most cases, promote the efficiency and economy of a plant's heating system.

That unit heaters offer a worthwhile opportunity to the engineer for reducing plant heating costs throughout the coming—and following—heating seasons is conclusively shown by the experiences of many plants. Numerous cases can be cited which show the specific economies or improved results obtained. To aid in visualizing the possibilities in modernization of this kind, several of them are summarized here.

Plant Experience with Unit Heaters

For instance, a survey of a foundry building by Samuel R. Lewis, Consulting Engineer, showed a possible reduction in the yearly heating cost of over $1,700 through replacement of the hot-water pipe-coil radiation with a modern unit-heater system. Walls, the columns, and the spaces between the columns overhead (save for the crane-ways) were covered with the coils, but at that

it was necessary to use coke-burning salamanders. Provisions for any temperature regulation were lacking. By keeping the heat down where needed, Lewis estimated that hot-water-type unit heaters would save the difference between 6,014,489 and 4,433,249 Btu per hr—at a unit cost of $1.10 per year per 1000 Btu's a worthwhile saving even neglecting that due to the elimination of the salamanders and the better results secured. Assuming the cost of the change to be $6,000.00, the following table shows the wisdom of the investment:

Investment	$6,000.00	
Interest on $6,000.00 @ 5%	300.00	
Saving in fuel		$1,739.00
Balance remaining in investment		4,621.00
END OF SECOND YEAR:		
Investment	$4,621.00	
Interest on $4,621.00 @ 5%	231.00	
Saving in fuel		1,739.00
Balance remaining in investment		3,113.00
END OF THIRD YEAR:		
Investment	$3,113.00	
Interest on $3,113.00 @ 5%	155.00	
Saving in fuel		1,739.00
Balance remaining in investment		1,529.00
END OF FOURTH YEAR:		
Investment	$1,529.00	
Interest on $1,529.00 @ 5%	76.00	
Saving in fuel		1,739.00
Investment all cleared with margin of	134.00	

A similar example has been reported by The Bishop & Babcock Sales Co., Cleveland, Ohio; 15,500 sq ft of steam radiation in a machine shop were replaced by nineteen unit heaters having a total capacity less than that of the radiation removed. The principal difficulty leading to the change was that, because of the height of the building, the heat delivered at the working zone was unsatisfactory, the cost excessive. There was no increase in fuel consumption after the units were installed, and considering that heating is now obtained where it was not before, it is clear that the improvement was justified economically.

In an article earlier this year, A. W. Moulder of Grinnell Company, Providence, R. I., cited two cases of unit-heater applications in different plants which proved to be advantageous. One is the plant of a hosiery manufacturer; replacing the old pipe coils with overhead-type unit heaters provided with thermostatic controls—and the installation of new vacuum pumps and a feedwater heater—made it unnecessary to provide more boiler capacity for an addition to the plant, as was at first contemplated. The other building—originally a cotton mill and later occupied by a tobacco concern—was equipped with units and thermostatic controls, "greatly reducing" the steam consumption.

A bulletin recently issued by Grinnell states that natural air circulation makes ceiling temperatures excessively high and that in the average factory 65 per cent of the heat-loss area is in roofs, monitor, or skylight glass. By using units which keep the heat down, fuel cost can be reduced. Several examples are quoted from this source:

A large plant was remodeled for the manufacture of airplane motors and the records now show a saving in steam consumption of 29 per cent by the use of unit heaters. Another example concerns a railway and dock shop installation, an old system being replaced by units. Coal consumption was reduced from 223 tons to 148 tons per season, a saving of 35 per cent. The third example refers to a large manufacturing concern operating more than fifty separate plants. Overall savings of 50 per cent are reported due to unit heaters; in one factory equipped with units and an automatic control system, savings of $50.00 a day are recorded.

Representative Installations of Units

The unit-heater installation in the plant of the Edward Katzinger Company, Chicago, shows how this modern method of industrial space heating meets the problems involved. This company manufactures kitchen utensils of various kinds, such as egg beaters, paring knives, etc. The plant—only a few years old—consists of a two-story section with a flat-slab roof and a one-story section with a saw-tooth roof. Two views are shown in Fig. 1.

In the two-story section it was necessary to counteract the large heat loss through the windows and walls and, on the second floor, through the roof. The single-story, saw-tooth roof section presented a special problem in that it was necessary to keep the heat away from the roof to promote economy and at the same time to heat rather large open floor areas. The heating requirements are met by twenty-five unit heaters (Ilg Electric Ventilating Co., Chicago), each with a rated capacity of 155,000 Btu or a total of 3,875,000. By proper location of the units, effective and economical heat distribution has been secured without using any floor space; the various bins of material, machines, and other obstructions do not interfere with distribution of the heat.

Four 259,000-Btu unit heaters are also used at this plant for the quick drying of enameled kitchen utensils, necessary to maintain the production rate. With the units, this drying is accomplished so that the production speed reached in other steps in the production process is maintained in the drying.

In Fig. 2 are illustrated several unit heaters (The Unit Heater & Cooler Co., Wausau, Wis.) installed in industrial plants; the units shown in *a* are placed 44 feet above the floor. The units pictured in this figure are designed to deliver the warm air to the working zone. These units are low-temperature, large-volume types; the steam chamber is of high-test iron around which is cast special alloy aluminum fin surface giving metal-to-metal contact and designed to withstand steam pressures up to 250 lb. The casing and louvres are built substantially to avoid their being distorted or bent.

An installation of turbine-driven unit heaters in a railroad shop is illustrated in the upper view of Fig. 3; where high-pressure steam is available units of this type often have advantages. As the exhaust steam from the turbine is used in the heater

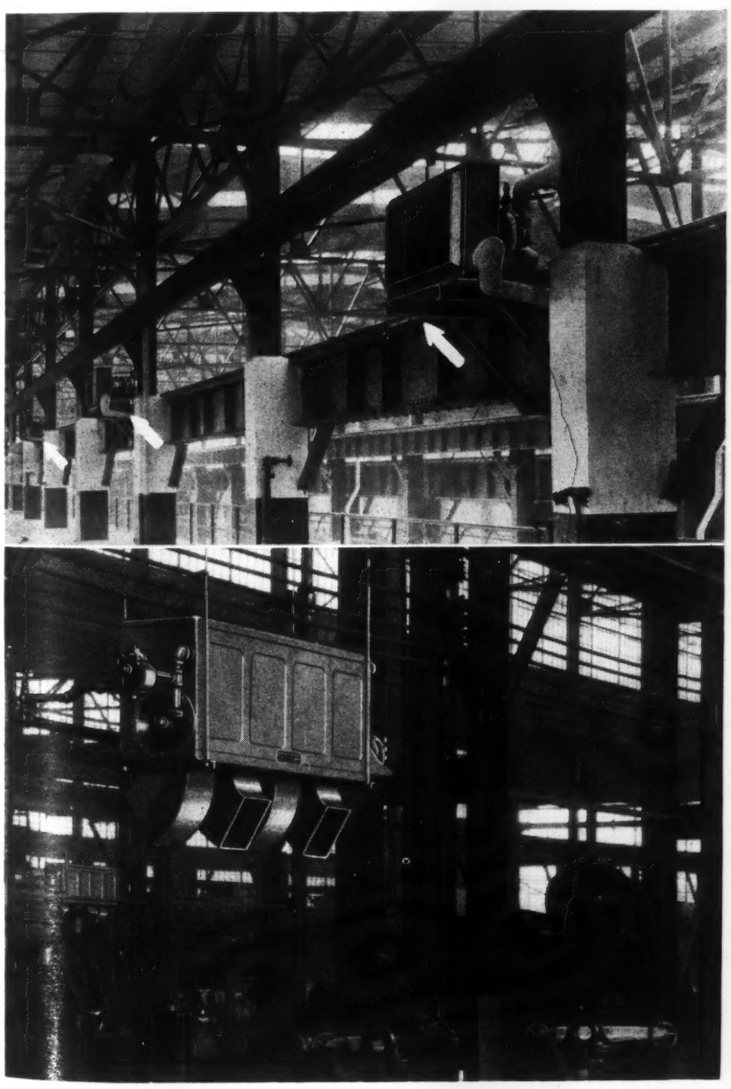
element an economical arrangement is provided and no electric wiring is necessary.

The lower part of Fig. 3 shows a unit installation in a foundry company plant. The type of unit shown can be suspended in an inverted position (as is the case here) or floor mounted, wall-bracket mounted, horizontally mounted, etc. Units of this general type are recommended for heating large areas, a small suspended-type propeller-fan unit also being available from the manufacturer (Buffalo Forge Company, Buffalo, N. Y.) where lower capacity is needed. Suspended and floor-type gas-fired unit heaters (also offered by the maker of these units) are becoming popular for many industrial applications, one recent shipment consisting of 64 of the suspended units.

A unit arranged for washing and humidifying the air is also available. A revolving drum of air washer, eliminator and scrubber plates is provided in the lower part. The drum is revolved by the air current and the plates continually dip into a water tank. Often a few of these units can be located advantageously near dust-producing machinery.

Another feature often desired is a unit with electric bypass damper control. With this control the unit is driven by a constant-speed motor providing full volume of air circulation at all times and giving minimum outlet temperatures. The controller actuates two interlocked dampers which allow recirculated air to enter above the heater element.

Fig. 3—Below: Unit-heater installation in a foundry company plant. Above: Turbine-driven units in a railroad shop



Use of Electric-Type Units

The adaptability of electric-heated units as a method of furnishing low-cost auxiliary heating in plants and buildings was described in an article by Lee P. Hynes published in the October, 1932, issue. Among the cases where this type of heating can be economically utilized, the author mentioned special locations in plants requiring heat when the plant as a whole does not; plants needing but little heat during the day because of large amounts of waste heat from lights, machinery, etc.; other cases where 24-hour operation of the regular fuel-burning heating plant would not be justified by the amount of heating needed.

An example of this use of such units in the press-room of a printing plant was cited. The building is heated with steam supplied by low-pressure boilers, the fires of which are banked at night. As the press-room needs heat 24 hours a day, two heavy-duty electric-type unit heaters were installed which have proved to be "very satisfactory." The heaters are equipped with thermostatic control. Another case is that of a paper-stock room requiring a moderate amount of heat at all times to overcome humidity conditions and protect the sprinkler system from freezing.

Recent Improvements in Unit Heaters

Unit heaters are constantly being developed and improved both in their construction features and in their application to plant heating. Recently, several manufacturers have announced improvements which should be of interest to prospective users. Others, it is understood, are working on new developments, in some cases extensive re-design being involved.

One concern (McCord Radiator & Mfg. Co. of Detroit) has made a number of changes in construction and design of its heater since the last heating season. The appearance has been improved by making a shell that is completely removable from the core. A new type of black baked finish also adds to the appearance. The spiral fin tubing is brazed into the heavy copper header. A special design of steam-inlet and outlet cast-iron tanks are removable and are held in place by large cap screws placed on 1-in.

(*To page 70, back Advertising Section*)

Include in $3,300,000,000 Public Works Program

The Modernization o

A PLAN for the expenditure of a portion of the $3,300,000,000 to be appropriated by the government for public works is presented here. It recommends that heating, power, plumbing, electrical and allied mechanical equipment in existing public buildings be modernized where such modernization will prove self-liquidating within a reasonable time.

The plan has been proposed by Russell Grinnell, President of Grinnell Co. There is probably not an engineer or contractor in this industry who can not testify to its soundness with examples of public buildings whose mechanical equipment can be modernized to advantage, or, equally conclusive, with examples of privately-owned buildings now enjoying the benefits and savings of modernized heating, piping and air-conditioning systems.

It deserves immediate consideration and action from the officials controlling the expenditure of the government's public works funds and of the states and municipalities which can avail themselves of these funds on such advantageous terms as they are being offered. It calls for the support of engineers, contractors, manufacturers and associations in this industry who can do both their government and their industry a real good by pointing out to the proper officials the business soundness of such expenditures.

This plan conforms with the aims of the government in its public-works program in all respects. In the first place, the government requires that expenditures be self-liquidating. The change, repair, addition, or replacement to heating, piping and air-conditioning equipment, properly engineered and directed, quickly pays for itself. Many examples to prove this are given in following columns.

In the second place, the government wishes by these expenditures to create employment and increase purchasing power. The kind of work proposed by this plan will provide much needed employment to a class and an industry which has been particularly hard hit by the depression. In proportion to the total cost it will call for a larger labor cost than road building and such projects now included in the government's plans

THE PLAN

[*Proposed by Russell Grinnell, President, Grinnell Co.*]

It is suggested that heating, power, plumbing, electrical and allied types of equipment in every existing public property be surveyed with a view to determining where modernization of such equipment will be self-liquidating within a reasonable time. Federal engineers, state engineers, city engineers and others who are responsible for such equipment are already aware of many places where economies can be effected by such modernization. Recommendations regarding many of these specific projects could, therefore, be made at once and others follow along at an early date. It is believed, however, that a survey of every property and a report regarding the efficiency of operation in the mechanical equipment with comparative results in one property as against other similar properties would reveal many more places where self-liquidating expenditures could be made than would otherwise be discovered.

It is suggested, therefore, that either an adequate force of engineers be placed at the disposal of the engineer now in charge of public properties or, in lieu of this, that the services of competent consulting engineers be employed to bring in such comprehensive reports.

Merely as a suggestion as to some of the items which might effect these economies, we list the following:

(*a*) *Installation of central heating plants for groups of public buildings which are now heated by separate boiler plants.*

(*b*) *Zoned automatic control of heating systems for large office buildings. This modern type of automatic control has shown savings of from 20 to 50 per cent in fuel bills in many large buildings.*

(*c*) *Installation of or elimination of power-generating equipment where estimated costs reflect favorably as against existing methods of generating electrical current or its purchase from a utility company.*

(*d*) *Modernization of piping with welded joints where present upkeep and maintenance is excessive.*

(*e*) *Installation of modern unit heaters or air conditioners which lend themselves to automatic control and economy of operation as compared to some of the old types of pipe coils and other heating equipment now installed.*

(*f*) *Insulation of piping or modernization of existing insulation for fuel saving.*

(*g*) *Installation of modern boiler equipment, elevator machinery, compressors, etc.*

It is believed, if those responsible for expenditure of public funds will carry on a comprehensive investigation along the general lines outlined above, they will establish these four facts:

1. *That considerable sums of money can be spent for mechanical equipment in a manner which will be quickly self-liquidating.*
2. *That the work of obtaining such comprehensive reports together with the engineering and supervision of such installations will provide much needed work for the so-called "white collar" engineering class of labor who by and large are probably more in need of immediate help than the average laboring man who works with his hands.*
3. *That work of this nature provides a great deal larger proportion of labor cost to total cost than does road building, reclamation projects, etc.*
4. *That such work does not add to productive capacity, but simply eliminates a present waste. Moreover, as weighed against new construction, it does not call for future fixed expense for maintenance, operation and depreciation.*

Mechanical Equipment

In Existing Properties

for expenditures. Furthermore, it will mean the utilization of local labor for these modernization jobs as well as increased employment in the factories making the mechanical equipment needed.

In the third place, the government does not want to add to productive capacity or to get into projects which will call for high future fixed expense. This plan does not add unneeded capacity and, instead of calling for future fixed expense, eliminates a present waste and cuts down future operating charges.

In short, it is the soundest kind of business and will be so recognized by the government, we feel sure, provided the facts are laid before the proper officials.

The suggestion that the services of competent consulting engineers be employed to survey and report on equipment in existing public buildings is in line with the general soundness of the plan. Trained in their profession, thoroughly familiar with the most modern design and equipment, experienced in work of this nature, they add to these qualifications the important one of quick action. They know their local conditions. In many cases they are already familiar with the mechanical systems of various public buildings in their communities. Immediately, therefore, they can get this plan operating.

For the same reasons the local heating, piping and air-conditioning contractor stands out as an effective medium through which this plan can be put into operation. In many cases he is the one who originally installed the systems in these buildings and he knows, therefore, the parts which can be improved to advantage with modern equipment developed since that time. Modernization is the type of work he has been doing these last few years in office buildings, hotels, hospitals, schools, institutions and industrial plants.

The plan speaks for itself. To those familiar with the savings to be effected through modernizing old mechanical equipment it is a "natural." Assembled here are typical examples of such modernization which will give those not so familiar an appreciation of the possibilities.

THE PROOF

FOLLOWING are examples of cost-cutting modernizing opportunities. Each is an actual case which reports how money was saved by rehabilitating mechanical equipment of a property, or tells what could be done in a specific instance toward reducing operating expense by a wisely-planned initial investment.

The few examples given here have been selected from the files or experience of a number of consulting engineers. Numerous others of course could be added, and a nation-wide survey would bring to light countless opportunities which would bear out the soundness of the plan shown on the opposite page.

Many Methods of Saving by Modernizing Mechanical Equipment

By **Samuel R. Lewis**
Consulting Engineer, Chicago

THERE are large public buildings in every community—public in the sense that every one uses them and sees them daily, even though the titles to all of them may not lie in the government. Such structures include churches, office buildings, museums, private schools, asylums, and colleges, as well as public schools, municipal and state eleemosynary institutions and the ever-present older buildings of the Treasury, the Postoffice, and the Judicial departments.

In nearly all of these buildings no repairs or improvements have been made for several years. They have just pulled along wastefully and often at the risk of such deterioration caused by failure in maintenance as to necessitate complete replacement. For instance, there is a surprisingly large number of tubular boilers with plugged-off tubes. When a tube in a fire-tube boiler begins to leak it is possible easily and quickly to plug it off at both boiler heads so that it fills with water and does no particular damage. However the boiler is thus deprived of a substantial fraction of its heat-absorbing surface, and its output will be reduced and the fuel consumption will be increased. In thousands of cases, rather than meet the investment for a new tube, the owner pays heavy interest and sinking fund increments on its cost by burning excessive fuel.

In an appallingly great proportion of cases when repairs or replacements are made the emergency is so violent that no intelligent consideration of a wise investment or reduction of operating cost is possible.

There is no money available to purchase new equipment and so by patching and by-passing efficiency-promoting apparatus an annual repair bill and operating cost is incurred which would liquidate in a short time the investment for adequate new machinery. This situation

especially is in evidence in public schools and in churches, which have received no increments to building funds for some time.

Since the depression commenced and since most existing buildings were designed, there have been many very radical improvements in the equipment for heating and ventilating and cooling. Despite the reduction in volume of sales, most progressive manufacturers have maintained their research appropriations and have improved their products.

Stokers Show Savings: For instance, automatic stokers for burning low-grade coal have been perfected to the point where (for existing apartment-type buildings especially) some manufacturers offer to accept all or nearly all compensation in installments as the stokers liquidate themselves by saving in operating cost.

In one municipal waterworks plant from May 1, 1930, to January 1, 1931, under hand firing 1,642 tons of coal were burned. From May 1, 1931, to January 1, 1932, with the same boilers but with electric-driven stokers, pumping more water rather than less, 1,065 tons of coal were burned. The saving in dollars per year by the change was (9,516-5,424) $4,092.00. The stokers cost the city $4,500. This investment, it will be observed, practically was self-liquidating in one year. There are hundreds of similar cases ready for attention.

In a Minnesota high school the annual saving due to stokers was $834.00. The investment was self-liquidating in two years.

A Colorado municipal electric-lighting plant spent $7,000.00 for a stoker, and saved $4,000.00 per year thereby, easily a two-year self-liquidation.

A large church saved in coal costs on a two-year average $775.00 per year. The stoker cost $1,600.00 or $1,680.00 with the first year's interest. $1,680.00—$775.00 leaves $905.00 or $960.00 with the second year's interest. $960.00—$775.00 leaves only $185.00 not liquidated at the end of the second year.

One Chicago apartment building saved $900.00 per year by purchasing a stoker which cost $1,100.00.

Almost every public school in the country burns coal, and does it manually, and is unable in most cases, due to legal restrictions, to make a contract for a stoker except upon advertisement, to the lowest responsible bidder, upon specific plans and specifications for a specific price. These restrictions might be lifted for a time so as to permit more latitude concerning repairs and replacements in existing school buildings.

Oil Firing: There are many public buildings which have inadequate boilers or too-small and too-low chimneys. Many of these could now be burning liquid fuel at a saving over present hand firing methods.

One large Chicago hospital changed from hand firing of coal to automatic firing with oil. The reason for the change was that coal delivery was noisy and disturbed the patients. During two years' experience it has been proved that oil fuel in this case rather unexpectedly is less costly than coal and that the $5,000.00 investment will be amortized in about eight years.

Gas for Water Heating: There is, especially in office buildings, a remarkable number of cases where coal or oil is being burned in big heating boilers during the summer merely to heat service water. Nearly all of these plants (especially in Chicago) could burn gas for this purpose during the summer and could pay for the necessary investment and engineering survey by the operating savings of one season only.

Service hot-water heating in many cases is one of our most potent sources of extravagance.

In one of the very finest Chicago cooperative apartment buildings the boilers and stokers operated all summer to furnish hot water and required an average of 2,100 lb of coal per day for this purpose. A very simple change in this equipment reduced the average daily consumption to 900 lb. The cost of the change was liquidated at the rate of 72 tons of coal, say $432.00, every four months. Such fabulous savings as this are extremely common and would affect potently the prosperity of the entire country if the owners of property knew about them and if they had the comparatively small capital necessary in each instance.

Temperature Control: There are many government buildings such as those which house Federal courts, postoffices and executive departments in which the heating and ventilating is from thirty to forty years obsolete. The old boilers have inadequate combustion-chamber volume, steam-heating systems have inefficient air venting, and ventilating systems have no air-cleaning devices. Most Federal buildings ignore entirely the effect of the daily travel of the sun around three sides of every building and therefore disclose wasteful open windows on the sunny sides. Almost all of these buildings, if provided with modern zoned automatic temperature control, would liquidate the investment for such apparatus in one or two seasons.

The Treasury department recently has realized the possibilities of economy when overheating is prevented and has installed zoned control and arrangements for reduced night temperatures, especially where central-station steam is purchased on a metered basis. It is true, of course, that where steam is generated on the premises waste would be prevented by proper control in equal proportions to the savings made where steam is purchased.

In one group of educational buildings in Indiana the installation of automatic temperature regulation represented an investment of $3,875.00. The first year after the control was installed a reduction of $3,308.00 in the cost of steam from that of the preceding year was accomplished. This improvement, including interest, was self-liquidating in less than 15 months.

In another case zoned control was applied to a skyscraper office building which has four-side exposure. On a month having an average outside temperature of 36 F in 1930 before the improvement was installed, the steam consumption was 9,500,000 lb. On a month having an average outside temperature of 36 F in 1933, after the improvement, the steam consumption was 7,165.000 lb. This saving of 2,335,000 lb of steam, expressed at 85 cents per thousand pounds, is equivalent to nearly $2,000.00 for that month. This investment would pay itself off in not more than two years. In many cases, however, adequate zone control of the type in

question costs so little as to be self-liquidating in one year. Studies of the records of seven similar buildings indicate that all of them show savings compatible with those of the case cited.

Air Cleaning: There are many public buildings, such as libraries and museums, which have been unable to maintain the air-cleaning facilities used with the ventilating plants up to the modern standard. The costs for cleaning and redecorating in order to maintain a halfway respectable appearance of walls and ceilings would purchase and maintain modern air filters in the air ducts, leaving as clear profit the reduction in damage to the books and exhibits.

Obsolete air-cleaning equipment in the ventilating system of one large Chicago bank building was replaced with viscous-coated automatic air filters. There was vastly improved cleaning of the air and a reduction in the annual operating cost of the air-cleaning apparatus of $3,500.00, which represented an annual return on the investment of over 35 per cent—an easy three-year liquidation.

Installation of air-cleaning apparatus on the intakes of large air compressors reduced the maintenance expense on these machines so much as to pay for the new equipment in one year and five months.

Ventilation and Exhaust Equipment: There are many manufacturing plants of corporations of the "AA1 class" which have approached obsolescence and rejuvenation of which would offer a large investment return. One such corporation has such a modernization process now underway, taking a bay at a time, and changing over its machine tools and conveyors and modernizing its exhaust systems for dust. This corporation happens to have enough cash reserve to finance the improvement. There are many manufacturing plants of unquestionable integrity and credit which could liquidate quickly a modernization investment provided someone would lend them the necessary capital.

In one specific case an old-time exhaust fan has required for many years 12.4 hp; say 9.3 kw per hour. It was replaced by a fan which the manufacturer developed during the last two years and which does the same work as the old fan, with equal effectiveness but with an expenditure of only 7.6 hp; say 5.7 kw per hour. The saving of 3.6 kw, on a basis of 24-hour days, 300 days per year, would buy the new equipment in one year only.

A downtown Chicago office building needed more rental area in the basement for a first-floor tenant, who, unless this space should be made available, would move across the street to a new building. A consulting engineer discovered that modern high-efficiency fans, motors and heat-transfer surfaces could be substituted for the existing ventilating apparatus. The modern equipment occupies so much less space than did the old apparatus that the needed additional basement offices became available. The modern equipment not only accomplished all that the old equipment accomplished, but it also ventilates the new basement offices adequately.

This investment by the building owner immediately became self-liquidating by the increased rental paid on a long-term contract by the tenant.

WE DO OUR PART

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HEATING, PIPING AND AIR CONDITIONING has complied with the President's agreement by conforming with the substituted provisions of the code submitted for the magazine and periodical industry. It has signed the required certificate of compliance and will hereafter display the blue eagle in each issue.

Savings in Operation of 50% and Higher Possible in Some Cases

By **Walter E. Gillham**
Consulting Engineer, Kansas City, Mo.

I HAVE not had the opportunity to check over any public buildings in Kansas City and vicinity but as many of our institutions are old and out-of-date they are no doubt very inefficient and much saving in operation could be made by modernizing them.

I have, however, checked over and have in my files reports on many private buildings, such as hotels and office buildings, which show that a saving of from 30 to 50 per cent could be made in operation if they were modernized. One example is a downtown office building, the income from which goes to help support a charitable institution. If this building were modernized, which could be done at a cost of $5,000 to $6,000, there would be a saving in operation of from $1,500 to $2,000. This would increase the income of the charitable organization, which would not have to draw so heavily on the allied charities for help.

Saves 70% First Year: Another example is a large downtown office building which is about twelve years old. This building was modernly equipped when erected with an up-to-date heating and ventilating system (including boiler plant) but regardless of this, steam was purchased for the heating of the building. The boiler plant was kept in shape and all ready for operation on a moment's notice. About two years ago the owners decided to operate their own plant continuously and their figures show that a saving of nearly 70 per cent was made in the first year's operation. Another example is that of a downtown hotel which was purchasing steam at a cost of about $5,000 per season; they installed their own plant with automatic stoker and claim a saving of nearly 50 per cent in operation by so doing.

Temperature Regulation: The boiler plant is not the only oppor-

tunity to make a saving in operation. There are many downtown buildings which have old one-pipe heating systems which if modernized with the installation of a two-pipe vacuum system with automatic temperature regulation, could save from 25 to 40 per cent in the operation of the plant.

I have interviewed many owners and building managers and they all recognize that these savings could, no doubt, be made but most of the buildings are now operating at a loss and they will not spend the money for these needed improvements. I do not know whether money for this private work would be available under the public works program or not but if it could be, it would provide much needed work for engineers, contractors, and material men if used for modernizing existing systems.

Plants Both Small and Large Show Opportunities for Modernization

By **Alfred Kellogg**
Consulting Engineer, Boston

THERE are many plants in buildings small and large that show savings due to rehabilitation. One conspicuous example follows:

Central Heating Plant Put in Shape: The central-heating plant of a state university provided steam at both high and low pressure through an underground conduit system to about twenty buildings within a radius of half a mile.

The writer was called upon, during the month of August of a recent year, by the business manager of the college, who stated that they had an installation of six 100-hp H. R. T. boilers, carrying 100-lb steam; that one boiler had been installed the previous summer, and that the six boilers were so overloaded that they felt that something must be done in order to carry them through the coming winter, as the state legislature (from which funds for a new plant were hoped for) would not meet for eighteen months.

The writer had never seen the plant. He made a one-day survey of the situation, and as the result, had all of the traps of every description, in all of the buildings, overhauled and made tight. There was installed an open-feed water heater and one new pump, with a few minor changes in boiler piping, and some excavation for pump pits of considerable size. These were the only changes recommended or made in the plant, the cost of which was about $8,500. The engineering fee amounted to $600.

Through the following winter, only four boilers were put in commission at any time, and there was effected a saving in coal of 1,500 tons, costing, delivered, a few cents above $10 a ton. This means that, with an investment of $9,000, a saving of $15,000 was made during the first winter.

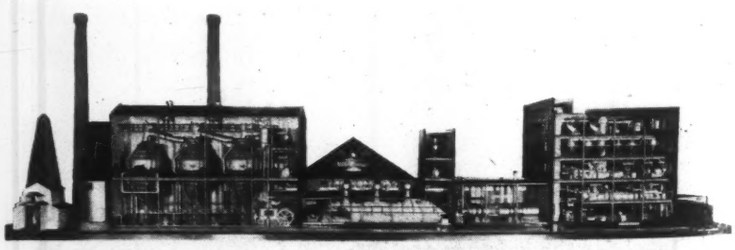
Needless to say, there was no need for rebuilding the boiler plant. with two spare boilers available.

World's Fair Plant Models Show Industrial Processes

THERE are a number of interesting and instructive plant models on display at A Century of Progress, Chicago World's Fair. Three of them which are of particular interest to readers of HEATING, PIPING AND AIR CONDITIONING are in the Agricultural building.

A model of the Atlas Brewing Company's plant complete in every detail shows the many piping systems in a modern brewery, the brew kettles, pumps, coolers, etc. A description of the piping services in the brewery of which this is a model was published in the April, 1933, issue. The National Biscuit Company has a plant on display which is unusually complete, even to showing the plant office with papers on the desks, typewriters in miniature, filing cabinets, etc. Much of the equipment is in apparent operation and the bakery process is clearly shown. The equipment for controlling temperature and humidity—so important in plants of this kind—is of course shown.

The modern production of salt is explained in an operating plant model by the Morton Salt Company; this model will later become a permanent exhibit in the Rosenwald Museum of Science and Industry at Chicago. It has been designed so that anyone inspecting it will start at the left, first viewing the piping and derrick construction of a salt well. Just to the right is illustrated a brine settling tank, which immediately precedes the huge, triple-effect vacuum evaporators. One of these evaporators is cut away, and shows clearly the steam belt and propeller which agitates the brine. The use of pipe lines indicates the flow from these vacuum evaporators to the washers and dehydrators. Immediately succeeding this building is one housing the open type evaporators, with hoods and cut-away sections showing the mechanical operations. The method of moving the salt over standard belt conveyors is shown from the evaporators to the rotary kiln dryers. The salt, after it leaves the dryers, is elevated to the top floor of the packaging department where it is carefully screened and dropped by gravity into storage bins from which it is fed to the various packaging machines.



Model of salt plant constructed to scale of one-half inch to foot at Morton Salt Company's exhibit at A Century of Progress

The Status of

Piping Standardization Today

By **Sabin Crocker***

SOME four years ago an article by the writer entitled "National Standardization of Piping"[1] appeared in this magazine. That article presented some of the reasons for national standardization, explained the *American Standards Association* set-up and procedure, gave a brief history of the more important piping standardization projects then in existence, and enumerated the piping standards already published, under way or contemplated. Much of the program planned at that time has since progressed to the point of release as tentative standards for criticism by industry, or, in some cases, has been finally accepted and published as American Standards. The present article, therefore, dealing with the present status of these projects, should be of interest and value.

THE STORY of national standardization of piping is told and interpreted by Sabin Crocker in this series of articles, of which this is the first.....The author knows his subject thoroughly, has been actively identified with piping standardization for the past ten years.....In addition to the complete list of piping standards which have been published and which are contemplated, the "reasons why" and other valuable comments are given.

Piping standardization projects can be conveniently divided into three broad classifications, which are

A. Dimensional standards
B. Materials specifications
C. Safety codes

The first two are mutually interdependent, so that one could not exist without the other being at least implied; the existence of the third hangs entirely on the other two. In view of this relation, it is convenient to discuss the status of piping standardization under the three topics named, and in the order indicated.

Dimensional Standards

Due to the existence of so large a number of dimensional standards, it is advisable for reference purposes to list those already published in one group arranged in the order in which they came out, listing those contemplated in a separate group. While this constitutes some repetition of the previous article, the convenience of having available a complete listing, coupled with the changes in status and additional projects which have developed in the meantime, seem to warrant such procedure. After completing the enumeration, including a brief descriptive note with each listing, more detailed explanation will be offered concerning certain items of particular interest.

A. S. A. Piping Standards Already Published.

Following is a list of Tentative Standards or American Standards already published. In addition to the SSP (steam service pressure) ratings given, most of the standards so designated have additional pressure ratings for non-shock hydraulic or other services. Until 1931 the steam ratings of American Standards for piping were designated as WSP (working steam pressure), but these initials were sometimes mistaken for "water service pressure." Acting on the recommendation of the *Manufacturers Standardization Society of the Valve and Fittings Industry*, the rating designations were changed to "steam service pressure," which is abbreviated SSP:

Pipe Thread, American Standard Serial No. B2-1919. Dimensions of the elements of the American (Briggs) standard taper pipe thread, and straight thread and lock-nut threads for pipe, formulas for calculating the dimensions of the threads; gage and working tolerances; methods of gaging the threads.

Screw Threads for Fire Hose Couplings, American Standard Serial No. B26-1925. Threaded parts of fire hose couplings, hydrant outlets, stand-pipe connections, siamese connections, and all other special fittings on fire line where fittings of 2½, 3, 3½, and 4½ inches nominal inside diameter are used.

Cast-Iron Screwed Fittings for Maximum SSP of 125 and 250 lb, American Tentative Standard Serial No. B16d-1927. Overall dimensions, metal thickness, beading, threading, markings.

Malleable-Iron Screwed Fittings for Maximum SSP of 150 lb, American Tentative Standard Serial No. B16c-1927. Overall dimensions, metal thickness, beading, threading, marking.

Cast-Iron Pipe Flanges and Flanged Fittings for Maximum SSP of 125 lb, American Standard Serial No. B16a-1928. Overall dimensions, metal thicknesses in flange and body, drilling templates, bolting, facing, marking, weights, and material requirements.

Cast-Iron Pipe Flanges and Flanged Fittings for Maximum SSP of 250 lb, American Standard Serial No. B16b-1928. Overall dimensions, metal thicknesses in flange and body, drilling

*Engineer, The Detroit Edison Company, and Member of Board of Consulting and Contributing Editors. Mr. Crocker is Secretary of *A.S.A.* Sectional Committee B36 on Standardization of Wrought Iron and Wrought Steel Pipe and Tubing, and of Subcommittee XXII of *A.S.T.M.* Committee A1, which is responsible for the preparation of piping specifications for high-temperature service. He also is a member of *A.S.A.* Sectional Committee B16 on Pipe Flanges and Fittings, and of Sectional Committee B31 on Code for Pressure Piping.

[1] HEATING, PIPING AND AIR CONDITIONING, June, 1929, p. 104.

templates, bolting, facing, marking, weights, and material requirements.

Steel Flanged Fittings and Companion Flanges for Maximum SSP of 100, 300, 400, 600, 900 and 1500 lb at a temperature of 750 F, American Standard Serial No. B16e-1932. The 100-lb standard is rated for 150 lb at 500 F, and 100 lb at 750 F. A table of pressure ratings for all the steel flange standards adjusted in 50-F steps for temperatures from 500 to 850 F is under consideration as explained later. Overall dimensions, including those of valves and steel flanged base fittings, metal thicknesses in flange and body, drilling templates, bolting, facings, marking, test pressures, and material requirements.

Cast-Iron Long-Turn Sprinkler Fittings, Screwed and Flanged for Maximum Hydraulic Working Pressures of 150 and 250 lb per sq inch gage, American Standard Serial No. B16g-1929. Overall dimensions, metal thicknesses in flange and body, threading, drilling templates, bolting, facing, marking, material requirements.

Cast-Iron Pipe Flanges and Flanged Fittings for Maximum SSP of 25 lb, American Tentative Standard Serial No. B16b2-1931. Overall dimensions, metal thicknesses in flange and body, drilling templates, bolting, facing, marking, material requirements, and weights.

Cast-Iron Pipe Flanges and Flanged Fittings for Maximum Non-shock Working Hydraulic Pressure of 800 lb, American Standard Serial No. B16b1-1931. Overall dimensions, metal thicknesses in flange and body, drilling templates, bolting, facing, marking, and material requirements.

Installation, Maintenance, and Use of Piping and Fittings for City Gas, Serial Number Z27-1933. Applies to the installation of gas piping in buildings, and the use of city gas, including natural, manufactured, and mixed.

Identification of Piping Systems, American Recommended Practice A13-1928. Identification of piping systems in industrial and power plants, which are not buried in the ground, with special reference to personal hazards in times of accident at a plant.

Wrench Head Bolts and Nuts and Wrench Openings. American Standard, Serial No. B18.2-1933. Dimensions of bolt heads and nuts for three series: American Standard Regular (1927 American Standard); American Standard Heavy (old U.S.S. dimensions used in piping work); American Standard Light (Automotive Practice). Table of open-end wrench openings for these three series.

Screw Threads for Bolts, Machine Screws, Nuts, and Commercially Tapped Holes. American Standard, Serial No. B1a-1924. Nomenclature of screw threads; form of threads; diameters and pitches of screws for various uses; classification of thread fits, tolerances, and allowances for threaded parts; and the gaging of threads.

A. S. A. Standards in Preparation.

The following piping standardization projects are now under way. Some have already been published in tentative form for comment.

Screw Threads for Hose Couplings, Serial No. B33- . Dimensions of screw threads for small hose couplings, ranging from ½-inch to 2-inch nominal size, intended for: garden hose; chemical hose; fire-protection hose other than chemical service; steam, water, and air hose with fine threads; and hose couplings with coarse threads.

Wrought-Iron and Wrought-Steel Pipe, Proposed American Tentative Standard Serial No. B36- . Outside diameters, wall thicknesses, weights, tolerances, and materials of wrought-iron and wrought-steel for several schedules of 1000 *P/S* values ranging from 10 to 160. The processes of manufacture covered include furnace-welded, forge welded, seamless, electric fusion-welded, electric resistance-welded, riveted, and lock-bar pipe.

Cast-Iron Flanged Ammonia Fittings, Serial No. B16h.

Center-to-Face Dimensions of Ferrous Flanged Valves, Serial No. B16. A partial schedule of center-to-face dimensions for cast-iron and steel flanged valves in the various pressure standards from 125 to 1500 lb SSP.

Malleable-Iron or Steel Unions (Brass Seat) for a Minimum Steam Pressure of 300 lb, Serial No. B16j.

Forged Steel Screwed Fittings and Steel Pipe Plugs. The preparation of standards for these articles is under consideration.

Cast-Iron Pipe and Special Castings, Serial No. A21-. Material, dimensions, pressure rating, methods of manufacture, elimination of unnecessary sizes and varieties, types of joint.

Plumbing Equipment, Serial No. A40. Materials, uniformity of roughing-in dimensions, efficiency of operation and other performance specifications. This sectional committee has recently issued a proposed standard for *Brass Fittings and Flared Copper Tubes, Serial No. A40-.* Overall dimensions, metal thicknesses, screw thread data, gaging data, and nut dimensions.

Proposed American Standard Steel Welding-Neck Flanges for minimum SSP of 100, 300, 400, 600, 900 and 1500 lb at 750 F. Overall dimensions, metal thickness of flange and hub, and facing. It is intended that this standard be added to the standard for Steel Flanged Fittings and Companion Flanges of which the material requirements, marking, test pressure, etc., will apply to the welding-neck flange standard.

Proposed American Standard Pipe Plugs of Cast Iron, Malleable Iron, Cast Steel, or Forged Steel for Use in Connection with Fittings Covered by American Standard 125-lb and 250-lb Cast-Iron Screwed Fittings and American Standard 150-lb Malleable-Iron Screwed Fittings. Overall dimensions, metal thickness, and threading for square head, bar or slotted head, and countersunk head patterns in nominal pipe sizes from ⅛ to 3½ inches.

Steel Flanges and Flanged Fittings for 2,500 and 3,500 lb SSP and Standardization of Blind Flanges. These proposed standards will contain information similar to that now included in the standard for steel flanged fittings and companion flanges and when completed, will probably form a part of it.

Brass and Bronze Pipe Fittings.

Bushings of Brass, Cast Iron, Malleable Iron, Cast and Forged Steel.

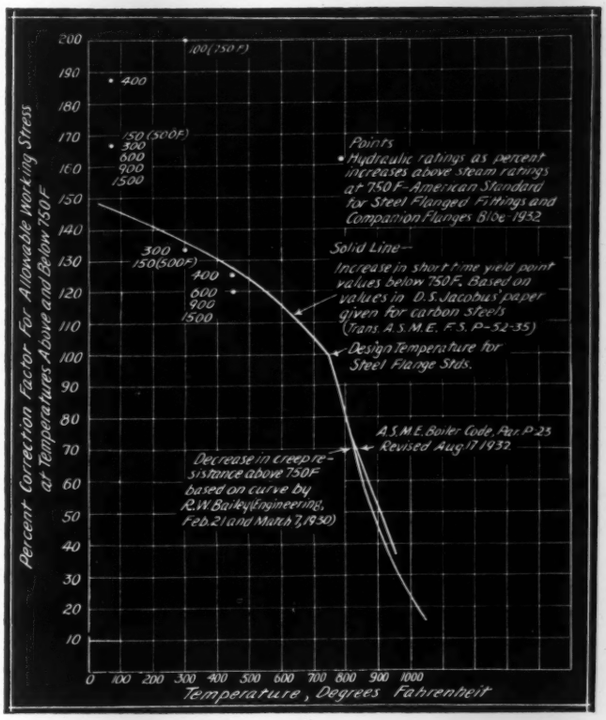
Many of the standards listed above have been in general use and are so well known that little need be said about them here. There are a number of points of interest, however, in connection with the newer standards and the proposed standards which merit further attention. This will be done under separate headings referring to individual standards.

Steel Flanged Standard

The American Standard for Steel Flanged Fittings and Companion Flanges was first published as a tentative standard in 1927, and re-published as an American Standard in 1932. The principal changes were: the addition of tables giving complete dimensions for companion flanges; the addition of base fittings; and a revision in the bolting requirements to call for American Standard Heavy Dimensions for Nuts which were included by Sectional Committee B18 in its new *American Standard B 18.2-1933 for Wrench Head Bolts and Nuts, and Wrench Openings.*

An interesting development now under consideration, and which probably will be approved for addition to the stand-

Fig. 1—Possible method for establishing allowable steam service pressure ratings for temperatures above and below the assumed design temperature of 750 F for carbon and low alloy steels


Percent Correction Factor For Allowable Working Stress at Temperatures Above and Below 750 F
100 (750 F)
400
150 (500 F)
300
600
900
1500
Points
Hydraulic ratings as percent increases above steam ratings at 750 F-American Standard for Steel Flanged Fittings and Companion Flanges B16a 1932
Solid Line—
Increase in short time yield point values below 750 F. Based on values in D. S. Jacobus' paper given for carbon steels (Trans. A. S. M. E. F. S. P-52-35)
Design Temperature for Steel Flange Stds.
A. S. M. E. Boiler Code, Par. P-23 Revised Aug. 17 1932
Decrease in creep resistance above 750 F based on curve by R. W. Bailey (Engineering, Feb. 21 and March 7, 1930)
Temperature, Degrees Fahrenheit

ard in the near future, is a table of adjusted steam-pressure ratings for temperatures above and below the basic temperature of 750 F. Consideration of such a table came about through a request from the *A. S. M. E.* boiler code committee for permission to use the steel flange standards at higher than the rated pressures at 750-F steam service in cases where the steam temperature was considerably below 750 F. This request was referred by Sectional Committee B16 to its Subcommittee 4 on Materials and Stresses, and to the *American Society for Testing Materials* for joint consideration. About the same time, Sectional Committee B36 on Pipe desired to have reduced pressure ratings set for pipe made to *A. S. T. M. Specification A106* when used at temperatures above 750 F, and made such a request to the *A. S. T. M.* In view of the complementary nature of these requests, it was decided to consider them as a single problem involving the properties of metals at elevated temperatures. As the materials in question were carbon steels not of a composition ordinarily used for extremely high temperatures, it was decided to work out a table of adjusted pressure ratings for temperatures from 500 to 850 F. To cover higher temperatures, the note previously in the *A. S. T. M.* specifications used in connection with the steel flange standards, which reads to the effect that "at the discretion of the designing engineer higher temperatures with appropriate pressures may be used," was retained.

In arriving at a reasonable schedule of adjusted pressure ratings at temperatures other than 750 F, a thorough study was made of published creep data and available experience in the use of the steel flanged standards at such temperatures. The boiler code committee had already published a schedule of reduced allowable working stresses for temperatures up to 950 F.[2] R. W. Bailey of the Metropolitan Vickers Co. (England) had pointed out a way for reconciling inconsistencies between the creep data of different investigators by expressing creep strength at a series of temperatures as percentages of the strength at some lower temperature.[3] Where creep data analyzed by Mr. Bailey's method was compared with the boiler code values, the results agreed most encouragingly. For temperatures below 750 F, experience with boiler-feed lines and 500-F service with the 150-lb steel flanged standard offered a satisfactorily reliable criterion for design. These data, which are plotted in Fig. 1, were deemed by the joint subcommittees working on the problem to be sufficient basis to establish safe working stresses at temperatures from 500 to 850 F.

Analysis of the formulas used in the design of pipe, fittings, flanges, and bolting used in connection with the steel flanged standard indicated that pressure ratings could be adjusted to temperature in identically the same manner as allowable stresses. Hence it was decided to recommend a table of adjusted service pressures at temperatures from 500 to 850 F based on the curves shown in Fig. 1. These adjusted pressure ratings are shown in Table 1.

[Future articles in this series will discuss wrought iron and wrought-steel pipe standards, dimensions of welded and seamless steel pipe, boiler-code pipe formulas, materials specifications, the proposed code for pressure piping, etc.—*Editor.*]

[2] *Mechanical Engineering*, May, 1932, p. 370.
[3] *Engineering*, February 21 and March 7, 1930.

Table 1—Adjusted Service Pressures for Power Piping at Temperatures Below and Above 750 F

Steam Service Pressure Rating at 750 F	100	300	400	600	900	1500
Service Temperature, Degrees F	Adjusted Service Pressures					
500	150	370	500	720	1080	1800
550	120	360	480	720	1080	1800
600	115	350	460	700	1040	1740
650	110	330	440	670	1000	1660
700	105	315	420	630	950	1580
750	100	300	400	600	900	1500
800	85	250	325	500	750	1250
850	70	200	270	400	600	1000

Study Course in Air Conditioning

In cooperation with the New York board of education the New York City heating, piping and air conditioning contractors' educational committee is offering, without charge, a course of study in air conditioning, refrigeration, and steam engineering. Registration will be September 18, 19, and 22, 7:30 to 9:00 p. m., at the Brooklyn evening technical school and the Stuyvesant evening trade school. Classes open September 25. The instructors are Harold L. Alt and John Everetts Jr.

Heat Loss from Copper Piping

Tables Show Transmission of Heat from Copper Pipes and Tubes ...Insulation, Lacquering Discussed

CONFUSION exists—to some extent—on the subject of heat loss from copper pipe and tubing and its insulation. Here is an article which, with the accompanying tables, puts the problem on an engineering basis.....The author is an authority on heat transmission and insulation, has been engaged in research along these lines for some years. Many of the tests which yielded the data given here were made especially for this article.

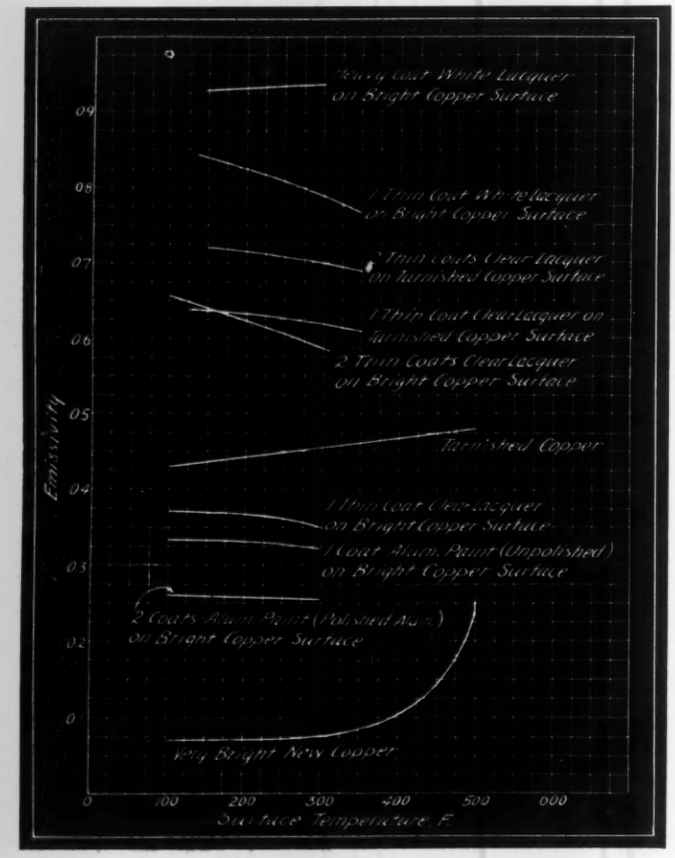


Fig. 1—Emissivity coefficients for bare and lacquered copper surfaces at various temperatures

By **R. H. Heilman***

COPPER tubing has become popular recently for conveying low-pressure steam, and hot and cold water. In some instances elaborate claims have been made as to the saving in heat by the use of copper pipes as against black steel pipes. Quite often copper pipes have been left uncovered, where the use of a suitable covering would have resulted in a very profitable investment for the owner. It is therefore the purpose of this paper to give the results of emissivity determinations made at Mellon Institute, from which the heat transmission from the surface of copper tubes and pipes has been calculated for various sizes and temperatures. From these determinations the heat transfer of copper and steel pipes can be compared.

Transmission of Heat from Copper Tube

Heat is transmitted from the surface of a copper tube to the surrounding air and walls by radiation and convection. The heat transmitted by free or natural convection is dependent upon the diameter of the tube, the temperature difference between the tube and the surrounding air, the average temperature of the tube and surrounding air and the position of the tube.

For horizontal tubes the transmission can be expressed by the formula

$$q_c = 1.016\ (1/D)^{0.2}\ (1/T_{avg.})^{0.181}\ \triangle t^{1.266}$$ Btu per sq ft per hr ..... (1)

where D = outside diameter of tube in inches.
$T_{avg.}$ = average temperature of tube and air, F absolute.
$\triangle t$ = temperature difference between tube and surrounding air, F.

The heat transmitted by radiation depends upon the nature of the surface of the tube and the difference between the fourth powers of the absolute temperatures of the tube and surrounding walls. The transmission by radiation can be expressed by the formula

$$q_r = 17.23 \times 10^{-10} \times e\ (T_1^4 - T_2^4)$$ Btu per sq ft per hr ..... (2)

where e = emissivity of surface.
T_1 = absolute temperature of tube, F.
T_2 = absolute temperature of surrounding walls, F.

From the above, it is seen that the heat loss by free convection is the same for copper and steel pipes for a

*Senior Industrial Fellow, Mellon Institute of Industrial Research, Pittsburgh, Pa.

given O. D. of pipe. However, the outer diameter of a copper tube of a given nominal pipe size is usually less than a steel pipe and the loss by convection per lineal foot of tube is therefore less than the convection loss from the same nominal size steel pipe. This difference is usually small, since it is proportional to the outside diameters of the two materials.

The main difference in the heat transfer between the copper and steel pipes is due to the decrease in radiation from the copper pipe. Since the radiation under given temperature conditions is directly proportional to the emissivity of the surface, it is seen that the lower the emissivity or the brighter the surface of the tube the lower will be the radiation loss. New copper pipe having a bright, more-or-less polished surface will therefore have a lower rate of heat transmission than a black steel pipe having a higher emissivity coefficient. If the copper tube is kept bright and clean and is not allowed to tarnish, the rate of heat transmission will be kept low. However, if the pipe becomes tarnished, the emissivity coefficient will increase and the heat loss by radiation will increase in direct proportion to the increase in the tarnish. New copper pipe will therefore transmit much less heat than old tarnished pipe. It is impossible to calculate accurately the heat transfer from copper pipes in the various stages of tarnish without knowing definitely the emissivity value at any given period. The tarnish is no doubt affected in many ways and in some instances the pipes may tarnish very rapidly.

Emissivity Value of Copper Determined

In order to have definite emissivity values for bright and tarnished copper, the writer has determined the emissivity of copper plate by means of a radiometer, which has been previously described.[1]

The emissivity values given in this paper are total emissivities, or for all the wave-lengths emitted at a given temperature. The emissivity of a surface can be defined as the ratio of the intensity of the radiation it emits, to the corresponding intensity of black body radiation at the same temperature. The emissivity of a "black body" is unity, whereas the emissivity of any other body is always less than unity.

The emissivity values at various copper temperatures were determined for new copper plate and for old copper plate. The new copper plate tested was the brightest that could be procured and the old copper plate was considerably tarnished, but still retained considerable reflecting power.

Discussion of Results

The results of the emissivity determinations are shown in Fig. 1. The heat losses from ½-in. to 6-in. diameter copper pipe of iron-pipe sizes have been calculated from the emissivity determination on the bright and tarnished copper plates by the use of Equations *1* and *2*. The results of these calculations are given in Table 1. Similar calculations have been made for ½-in. to 6-in. diameter copper tubes and the results are shown in Table 2. For comparison, the losses from horizontal bare iron or steel pipes are given in Table 3.

From the results of the emissivity determinations, it would seem that the heat losses as calculated for the new copper plate would be an absolute minimum and would probably never be obtained in practice as the new copper

[1] "Surface Heat Transmission," by R. H. Heilman. *Trans. Amer. Soc. Mech. Eng., FSP51, 287 (1929).*

Table 1—Heat Losses From Horizontal Bare Copper Pipe

(In Btu per square foot of pipe surface per hour. Air temperature 70 F)

Pipe Size, Inches	Actual O. D. Inches	Lineal Foot Factor[c]	Temperature Difference Between Pipe Surface and Surrounding Air													
			50 F		100 F		150 F		200 F		250 F		300 F		350 F	
			B[a]	T[b]	B	T	B	T	B	T	B	T	B	T	B	T
½	0.84	0.220	51.8	73.0	123	173	205	291	295	428	396	594	508	759	639	957
¾	1.05	0.275	49.7	70.9	118	168	197	283	283	416	378	576	489	740	616	934
1	1.315	0.344	47.8	69.0	113	163	187	275	272	405	363	561	471	722	594	912
1¼	1.660	0.435	45.8	67.0	109	158	181	267	261	394	349	547	452	703	571	889
1½	1.900	0.498	44.7	65.9	106	156	177	263	254	388	340	538	442	693	559	877
2	2.375	0.622	42.9	64.1	102	152	170	256	244	378	327	525	425	676	539	857
2½	2.875	0.753	41.5	62.7	98.3	148	164	250	236	370	316	514	412	663	523	841
3	3.500	0.917	40.0	61.2	95.0	145	158	245	228	362	306	504	399	650	507	825
3½	4.000	1.047	39.1	60.3	92.6	142	154	241	223	356	299	497	390	641	496	814
4	4.500	1.179	38.3	59.5	90.8	141	151	238	218	352	293	491	382	633	487	805
4½	5.000	1.310	37.6	58.8	89.1	139	148	235	214	348	287	485	376	627	479	797
5	5.563	1.458	36.9	58.1	87.4	137	146	232	210	344	282	480	369	620	471	789
6	6.625	1.736	35.8	57.0	84.7	135	141	228	204	336	274	472	359	610	459	777

[a]Bright. [b]Tarnished. [c]To secure losses per lineal foot, multiply square foot losses in table by this factor.

Table 2—Heat Losses From Horizontal Bare Copper Tubes

(In Btu per square foot of tube surface per hour. Air temperature 70 F)

Tube Size, Inches	Actual O. D. Inches	Lineal Foot Factor[c]	Temperature Difference Between Tube Surface and Surrounding Air													
			50 F		100 F		150 F		200 F		250 F		300 F		350 F	
			B[a]	T[b]	B	T	B	T	B	T	B	T	B	T	B	T
½	0.625	0.164	54.7	75.9	130	180	216	303	311	444	415	613	535	786	672	990
¾	0.875	0.229	51.5	72.7	122	172	203	290	293	426	391	589	505	756	635	953
1	1.125	0.293	49.1	70.3	117	167	194	281	280	413	373	571	483	734	609	927
1¼	1.375	0.360	47.4	68.6	112	162	187	274	270	403	360	558	467	718	589	907
1½	1.625	0.426	45.9	67.1	109	159	182	268	262	395	350	548	454	705	573	891
2	2.125	0.556	43.8	64.0	104	154	173	260	249	383	334	532	433	684	549	867
2½	2.625	0.687	42.1	63.3	100	150	167	253	240	374	322	520	418	669	530	848
3	3.125	0.818	40.8	62.0	96.7	147	161	248	232	366	311	509	405	656	515	833
3½	3.625	0.949	39.8	61.0	94.3	144	157	244	227	360	304	502	396	647	504	822
4	4.125	1.079	38.9	60.1	92.2	142	154	240	222	355	297	495	388	639	494	812
5	5.125	1.341	37.4	58.6	88.7	139	148	234	213	347	286	484	374	625	482	795
6	6.125	1.603	36.2	57.4	85.9	136	143	230	207	340	276	476	363	614	464	782

[a]Bright. [b]Tarnished. [c]To secure losses per lineal foot, multiply square foot losses in table by this factor.

Table 3—Heat Losses From Horizontal Bare Iron Pipes

(In Btu per square foot of pipe surface per hour. Air temperature 70 F)

Pipe Size, Inches	Actual O. D. Inches	Lineal Foot Factor[c]	Temperature Difference Between Pipe Surface and Surrounding Air						
			50 F	100 F	150 F	200 F	250 F	300 F	350 F
½	0.84	0.220	124	270	443	642	877	1152	1470
¾	1.05	0.275	118	257	420	614	840	1104	1414
1	1.315	0.344	114	250	410	600	822	1080	1382
1¼	1.660	0.435	112	245	402	588	808	1062	1361
1½	1.900	0.498	110	241	396	580	797	1050	1348
2	2.375	0.622	107	235	387	570	785	1035	1327
2½	2.875	0.753	104	229	378	556	765	1011	1298
3	3.500	0.917	102	225	371	546	750	993	1291
3½	4.000	1.047	101	222	365	538	743	981	1263
4	4.500	1.179	100	219	360	532	735	972	1253
4½	5.000	1.310	99	217	357	526	730	966	1245
5	5.563	1.458	98	215	354	522	725	960	1239
6	6.625	1.736	97	213	351	518	720	954	1232

[c]To secure losses per lineal foot, multiply square foot losses in table by this factor.

plate as received had an emissivity value actually lower than was previously obtained for a polished brass plate under the same temperature conditions. And since the plate tested was especially selected for brightness, it is reasonable to expect that the emissivity values obtained would be somewhat lower than would ordinarily be obtained for clean new copper tubes or pipes.

Comparison with Other Tests

It is desirable to compare the tests of Willard and Kratz[2] on clean new sheet copper and slightly dull copper, and the tests of Houghten and Gutberlet[3] on new copper as received, with those of the writer. The tests by Willard and Kratz were performed on cylinders 10 in. in diameter made up from No. 28 gage copper and heated to 215 F with steam. The tests by Houghten and Gutberlet were performed on, as received, copper tubes ¾-in. to 4-in. in diameter, with steam at 212.3 F to 213.5 F. Since the writer's tests were conducted over a considerable temperature range, the emissivity at 213-215 F can be taken from Fig. 1. Tests by the above observers were also made on iron surfaces, the emissivity values of which were more accurately known. Heat losses obtained from the iron surface tests checked very satisfactorily with Equations *1* and *2* developed by the writer, and emissivity values calculated from the tests on copper surfaces by means of the same equations should, therefore, be fairly accurate. The results of the actual tests and the calculations are shown in Table 4.

Comparing the emissivities, it is seen that the ¾ and 1-in. new copper tubes had emissivity values as received of 0.50 and 0.48 respectively or higher than the tarnished plate whose emissivity under the same temperature conditions was 0.44. The emissivity of the 1¼-in. tube was slightly lower, and that of the 4-in. tube was considerably lower than the tarnished plate but almost four times higher than the bright copper plate whose emissivity was 0.07 at 213 F. The slightly dull 10-in. diameter sheet copper cylinder had an emissivity value approximately 20 per cent higher than the tarnished plate, while the clean new sheet copper and the re-cleaned like new sheet copper in the 10-in. cylinder had emissivity values slightly lower than the tarnished copper plate. With the exception of the 4-in. tube, all the tests on the new copper surfaces indicate emissivities close to that of the tarnished plate. If the values of U (or the combined heat transmitted by radiation and convection in Btu per hr per sq ft per F) are compared with the calculated U values, based on an emissivity value of 0.44, the results are even closer, as would be expected since the transmission by the two methods are combined.

[2]Journal Section of the *A. S. H. V. E.*, Heating, Piping and Air Conditioning, February, 1931, page 141.
[3]Journal Section of the *A. S. H. V. E.*, Heating, Piping and Air Conditioning, January, 1932, page 47.

Summary

In any practical case the total transmission by radiation and convection has to be considered and from the results of the above it can be reasonably concluded that in most practical cases the heat loss from new clean copper pipes and tubes will be close to those shown in Tables 1 and 2 for tarnished copper, even where care is taken to keep the copper pipe absolutely clean. In a great many cases the pipe is soiled with pipe compound, handling with soiled gloves, etc. If this material is not removed from the pipes, the heat losses will be greatly increased. It is believed that after the pipes have been in service for some time and have become considerably tarnished that the losses will be somewhat higher than those shown in the tables.

It should also be borne in mind that the effect of the emissivity of the pipe becomes increasingly important as the diameter and the temperature increase, as the convection per square foot of surface decreases as the diameter increases while the radiation loss per square foot remains the same, and the radiation increases with temperature more rapidly than the convection.

Lacquering of Copper Pipes to Prevent Tarnishing Increases Heat Loss

The above discussion points out some of the variations in heat loss that might be expected with copper pipe and indicates to some extent why there is some confusion as to the actual value of copper piping in preventing heat loss.

In a recent article in another magazine the author quotes a contractor writing about a job as follows:

"The mains installed in the concrete pipe conduits and under the concrete floors were covered with three-ply air cell sectional insulation. All other pipe in the boiler room (including the steam header) was cleaned off, polished and given a coat of white lacquer.

"In our experience, we have found that the necessity for covering depends upon conditions entering into the installation. We find that in boiler rooms where copper piping is polished and lacquered to retain its bright surface, there is little evidence of radiation and heat loss. In fact, we should say that the results are slightly better than when three-ply air cell insulation and black iron pipe is used.

"What would happen to uncovered piping which is allowed to accumulate dust and become rough is a matter outside of our experience. The writer believes that with copper piping prepared in some ways to retain its high polish, or, if exposed, so that it can be wiped clean periodically, covering is unnecessary. If, however, copper piping is concealed where the above is impossible, we feel the covering is necessary."

It is difficult to reconcile the statement, "that when a copper pipe is polished and given a coat of white lacquer that there is little evidence of radiation and heat loss," with the fact conclusively proved by numerous investigators that, when a copper or any other pipe is coated with a lacquer of any color, the heat transmitting properties are immediately increased to practically the same as that of a black iron or steel pipe.

Tests conducted by the writer in 1929 showed that, at a temperature of 200 F, both white and black lacquer had an emissivity value of 0.95, while that of a steel plate was exactly the same. At a temperature of 100 F, the black and white lacquer had an emissivity of 0.80. The increase at 200 F was due to the fact that at a temperature of 200 F the lacquer had lost considerable of its lustre and was therefore emitting radiant heat at nearly as high a rate as a perfectly black body.

Since the steam header in the installation referred to by the above-quoted contractor was given a coat of white lacquer the minimum temperature of the lacquered pipe would be 212 F and the steam header would, therefore, be emitting heat at the same rate as a black iron pipe would under the same steam temperature. The size of the steam header is not mentioned, but if we take a 2-in. pipe (which is probably considerably smaller than the header installed), for example, we find by interpolation from Table 3 for 142-F temperature difference that the heat loss from the white lacquered surface is 363 Btu per sq ft of surface per hr. The heat loss through three-ply air cell on a 2-in. pipe under the same conditions would be approximately 113 Btu per sq ft per hr, a saving of 250 Btu per sq ft per hr, or an efficiency of 69 per cent, which certainly is worth while, especially where a considerable amount of piping is involved.

If the piping had not been lacquered, the loss from the bare copper pipe would have been considerably less, the loss for a 2-in. pipe when first installed, using an emissivity value of 0.44, being by interpolation 239 Btu per sq ft per hr. As mentioned before, an emissivity value of 0.44 is approximately the average value that might be expected for new copper pipes as installed. For the new copper pipes as installed, the use of three-ply air cell insulation would cut the loss down to less than one-half of the loss from the bare copper pipe.

If the copper pipes had been lacquered with clear lacquer, the heat losses would be less than for the white lacquer but still considerably higher than for the unlacquered surface.

Tests Made on Lacquering Copper

Fig. 1 shows the results of tests conducted by the writer on various lacquers and aluminum paint on copper surfaces. The 1929 tests of the white lacquer were again determined with another lacquer and substantially the same results were obtained. At 100 F, the emissivity was 0.79, while at only 150 F the value had increased to 0.93. Various tests were conducted with clear lacquer as former tests had indicated that on bright surfaces the use of very thin coats resulted in a much lower emissivity value than was obtained for the thicker coats. This is no doubt due to radiant energy being emitted from both the copper and the lacquered surfaces when a very thin coat of lacquer is applied. It was also noticed that, with the thin coats, the emissivity value decreased as the temperature of the surface increased, which was not the case with the bare metal or the thicker coat of white lacquer. The decrease is no doubt due to the fact that at the higher temperatures the wave lengths most strongly emitted become shorter and can therefore more readily penetrate the very thin lacquer film. This phenomenon might be utilized for the measurement of film thickness or the spacing or particle size of materials composing the film.

In the first tests, the clear lacquer was applied to the

Table 4—Results of Tests on Emissivity of Copper Surfaces

Source	Material	Size, In.	u	u'	e	e'
Houghten & Gutberlet	New copper as received	¾	2.01	1.91	0.50	0.44
Houghten & Gutberlet	New copper as received	1	1.91	1.85	0.48	0.44
Houghten & Gutberlet	New copper as received	1¼	1.76	1.80	0.41	0.44
Houghten & Gutberlet	New copper as received	4	1.32	1.58	0.27	0.44
Willard & Kratz	Sheet copper, slightly dull, not corroded	10	1.58	1.45	0.53	0.44
Willard & Kratz	Sheet copper, re-cleaned-like new	10	1.39	1.45	0.40	0.44
Willard & Kratz	Sheet copper, clean new surface	10	1.41	1.45	0.42	0.44

u = Btu per hour per square foot per F steam to air, test results.
u' = Btu per hour per square foot per F steam to air, calculated from Heilman equations where $e = 0.44$.
e = Emissivity of copper surface calculated from Heilman equations.
e' = Emissivity of tarnished copper, Heilman tests.

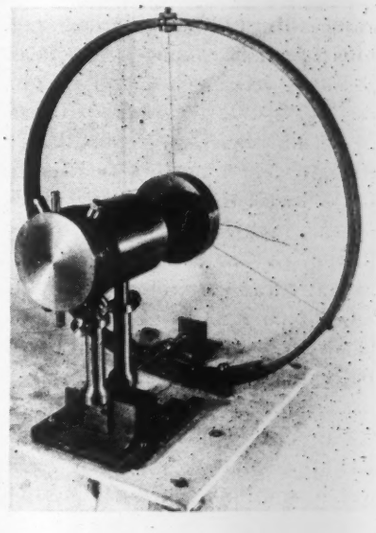


Fig. 2—Thermopile used in the investigation

very bright copper plate with a small camel's-hair brush so as to obtain as thin a film as possible. One extremely thin film increased the emissivity value at 150 F from 0.07 to 0.37 or more than five-fold. Two thin coats increased the value to 0.64 or more than nine times that of the bright copper. When the tarnished copper plate was coated with one thin coat of clear lacquer, the emissivity value was increased from 0.43 to 0.64 and two thin coats increased the value to 0.84.

It is advisable to paint small copper or iron pipes such as ½ in., which are often not insulated, with aluminum paint made from polished aluminum powder. This gives an emissivity value considerably lower than the average of new copper pipe and is more or less permanent; the small pipes in the basement of the writer's residence have been painted for several years and the aluminum paint appears to be practically the same as when first applied. The use of aluminum paint places iron pipes on the same basis as copper in regard to heat transmission. The aluminum paint is advocated only for the small pipes as the total surface area is usually small, while the convection and the radiation from the larger pipes is usually great enough to warrant the use of heat insulation, even for the lower temperatures used in hot-water heating systems.

Economical Insulation Thickness Same for Copper or Iron Pipe

Many engineers are under the impression that the economical thickness of heat insulation for copper pipes is less than that for iron pipes. This is not the case where copper pipe of iron-pipe size is used, for when the insulation is applied to the copper pipe the lower radiating effect of the copper surface—purely a property of the surface—is eliminated and the loss through the insulation is the same as when applied to the iron pipe.

However, the use of copper pipe will, for low-temperature conditions or for very low values of heat, give a saving over a bare iron pipe which would obviate the use of insulation. The conditions where this occurs would be determined from the value of the heat loss, operating conditions, etc., as there are conditions where even with iron pipe it does not pay to insulate. The use of copper pipe would, therefore, allow for a higher temperature or higher fuel cost before insulation would be required. When the temperatures and fuel costs are high enough that a worthwhile saving can be obtained by the use of the thinnest covering procurable over that of the bare copper, then the economical thickness does not have any relation to the nature of the pipe as the economical thickness is determined by that thickness of insulation which gives the lowest annual operating cost or by the last increment of thickness of insulation which will return, say, 15 per cent on the investment and, in this case, the loss from the bare pipe does not enter into the problem.

Where copper tubing is used having a diameter less than the nominal iron-pipe diameter, the heat loss through a given thickness of insulation will be less per square foot of surface for the iron surface but will be less per lineal foot of copper surface. For instance the loss per lineal foot of 1.125-in. O. D. tube covered with 1-in. thick insulation will be roughly 10 per cent less than the loss per lineal foot of standard iron pipe with the same thickness of insulation. As the diameter increases the ratio of heat loss greatly decreases so that for the larger sizes there is little difference in heat loss between the insulated iron pipe sizes and the insulated copper tube sizes.

For cold-water lines below room temperatures or where the heat is flowing into instead of out of the pipe, the problem is largely the prevention of sweating and frosting. In this case, insulation is always necessary and it is also advisable to use the same thickness for copper pipes or tubes that is recommended for iron pipes.

For most practical cases in which either copper or iron is used, a 1-in. thickness of insulation is recommended for cold-water lines and for low-pressure steam lines.

U. S. Civil Service Examinations

The United States Civil Service Commission announces the following-named open competitive examinations: Senior Engineer, Engineer, Associate Engineer, Assistant Engineer. Applications must be on file with the U. S. Civil Service Commission at Washington, D. C., not later than September 28, 1933. The examinations are to fill vacancies occurring in the Federal classified service throughout the United States. Entrance salaries range from $2,600 to $5,400 a year, less a deduction of not to exceed 15 per cent as a measure of economy and a retirement deduction of 3½ per cent.

Optional branches are aeronautical, agricultural, civil, construction, electrical, heating and ventilating, highway, mechanical, mining, radio, structural, and telephone engineering. Competitors will not be required to report for a written examination, but will be rated on their education and experience. Applicants must have had certain specified education and experience in engineering.

Full information may be obtained from the Secretary of the United States Civil Service Board of Examiners at the post office or customhouse in any city, or from the United States Civil Service Commission, Washington.

Handbook of Chemistry and Physics

Some 1700 pages of tabular data on chemistry, physics, and allied sciences are contained in the seventeenth edition of the "Handbook of Chemistry and Physics" published by the Chemical Rubber Publishing Co., Cleveland, Ohio. Among the new tables in this edition are integrals and interest tables in the mathematical section. The sections devoted to heat of formation and solution and vapor pressure have been revised. A table showing the variation of vapor pressure with temperature has been added.

This standard handbook is divided into a number of sections, including heat, properties of matter, hygrometric and barometric, sound, and others. Price of the deluxe edition is $6.00.

Fig. 1—The 22 East 40th St. building of the New York Trust Company, several floors of which were modernized with air conditioning to accommodate the general headquarters of the Johns-Manville Company

Conditions Air for Six-Floor Tenant

Office Building Modernized With Air-Conditioning System

[*By* M. O'Connell*]

THE MODERN office building provides conditioned air for the comfort and efficiency of its occupants. Existing buildings can be modernized to advantage with air conditioning.....An excellent example is the job described here, where several floors of a two-year-old building were equipped to furnish conditioned air for a tenant.

TRUE as it was in more prosperous times, it is even more so today—that high rentals go to the most modern office buildings; buildings that cater to the comfort of the occupants. They are characterized in general by acoustical wall, floor, and ceiling treatment, by high wattage indirect lighting, and air conditioning. Though air conditioning is the most recent of office improvements to gain general attention, it goes hand in hand with the others. No office space can boast of its sound-absorbing construction or its shadow- and glare-free lighting, if in the face of these innovations, its occupants swelter in a summer heat or stir restlessly in discomfort in the winter dryness of overheated rooms.

Air conditioning need not be built into a building as part of the original construction. It may be added later to the completed building, to the old building, with almost as little difficulty as any other form of "modernization." It is true that (aside from first cost and operating costs) the selection of air-conditioning equipment does involve such considerations as operating noise, space requirements, and flexibility, all complicated in the case of an existing building by a further consideration—the extent of building alterations necessary to accommodate the installation. However the problem is a "paper problem," and once it has been solved, the installation may be accomplished with ease and economy. As there are many types of office layouts, there are many combinations of refrigeration, air distribution, and control equipment to make many types of air-conditioning systems. And there is always the right one for the job.

Factors Considered on the Job

The installation at the 22 East 40th St. building of the New York Trust Company is an excellent example of modernization with air conditioning. It was made in a building over two years old at the time. Steam-jet refrigeration was selected because the building was already supplied with district steam for heating, and steam was available in summer at such low rates as to make the operating cost lower than with other types. The refrigeration equipment had to be located in the basement adjoining bank safety-deposit rooms and therefore had to be quiet. Steam refrigeration is relatively noiseless. It is vibrationless and the rushing sound which characterizes it is easily lagged and will not travel through building construction.

A cooling tower was decided on as a condenser-water economy measure. It could not be located on the roof without expensive reinforcing of the roof floor to support its weight, and as the only available roof area is forty stories above the basement much cutting and patching would have been required for the condenser-water piping and excessive water static pressures would result in the pipe lines and condenser. The cooling tower was therefore located in the basement and is a forced-draft type discharging into a convenient shaft; the shaft is

* Engineer, Sturtevant Cooling and Air Conditioning Corporation, New York City.

one of those lucky breaks which must be utilized to the utmost on a modernization job.

A booster-fan system of air distribution in conditioned areas was chosen because such a system permits of floor-by-floor control of conditions as well as exposure control, offering desirable flexibility. It makes for a

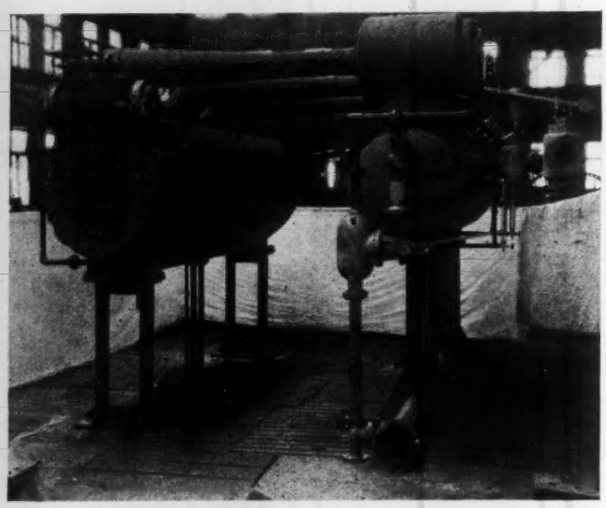


Fig. 2—Steam-jet refrigeration unit assembled in the shop before installation at the building. The plant has a present refrigeration capacity of 180 tons, a future capacity of 250 tons

small central dehumidifier unit and the breaking up of necessary fan-room space into small, easily-located units. A corridor return system was decided on to avoid use of a return duct system, saving floor space and head room which would otherwise be sacrificed to the risers and horizontal runs of such a system. Fortunately the arrangement of offices along a corridor lent itself to the scheme. Not all layouts are so accommodating.

This system was installed in conjunction with extensive alterations made to prepare six of the eight floors it treats for occupancy by the Johns-Manville Company as general headquarters. The space is made up chiefly of small private offices, but also includes many large clerical areas. Ceiling acoustical treatment and movable transite partitions arranged in many decorative schemes are features of the modern office layout.

Dehumidifier Room on Seventh Floor

The steam-jet refrigeration equipment is located in the basement and consists of the evaporator, surface condenser, three thermal compressors or jets, condensate pump, chilled-water pump, and inter-connecting piping and control. One of the present jets (70 tons capacity) is under automatic control by a positive open or closed action of the steam valve controlled from the temperature of water leaving the evaporator. The chilled-water pump supplies cold water from the evaporator to the dehumidifier room on the 7th floor, directly above the refrigeration equipment, where the water either passes through a mixing valve to the dehumidifier-pump suction or is bypassed and returned directly to the evaporator. The chilled water which enters the dehumidifier-pump suction to pass through the dehumidifier sprays returns to the evaporator through the dehumidifier overflow.

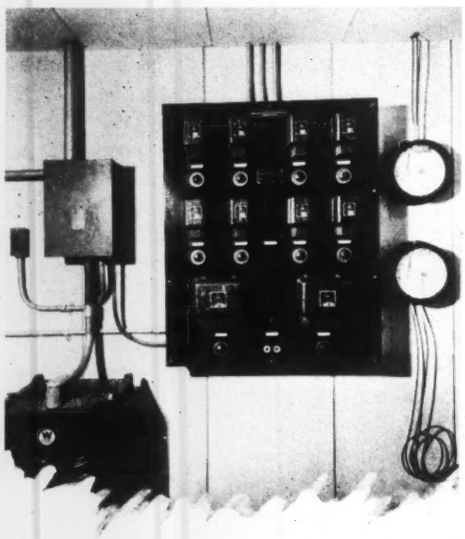


Fig. 3—The electric automatic control panel for the air-conditioning system

In the dehumidifier room there are, besides the dehumidifier and dehumidifier pump, a fresh-air intake, tempering heaters, and the main supply fan. The supply fan draws an automatically-controlled mixture of outside air and return air from the conditioned floors through the dehumidifier, and distributes it through a riser to booster-fan units, one on each floor.

One Booster-Fan Unit on Each Floor

Each booster-fan unit is equipped with filters and reheaters and is located in a small room which serves as a plenum chamber, where air from the main supply fan and return air from the respective floor mix before entering the fan through the filters. The bypassing of the dehumidifier is thus accomplished at the booster fan. Location of the filters in the booster unit insures a complete cleaning of each room air change.

Distribution of the conditioned air on each floor is made through duct work well concealed in the ceiling and partition treatment. Sound absorbers are fitted into the supply system to insure against the passage of fan-room noise through the ducts to the offices. Return of air to the booster fan room is through the corridors, the office doors being equipped with louvres, when necessary, to permit free exhaust of air from the offices to the passageways.

Cooling Tower in Basement

The "cooling tower" is located in the basement adjoining the refrigeration equipment. It consists of a two-bank, spray-type air washer, a circulating pump, and a double-inlet fan, in blow-through arrangement, discharging into a shaft which rises the full forty-story height of the building. This shaft is formed by an adjoining building serving as a fourth wall to the escape or fire shaft of the building in which the system is located. Air for the cooling tower is drawn through a sidewalk grating and passes freely through the basement to the open intakes of the cooling-tower fan. The cooling-tower

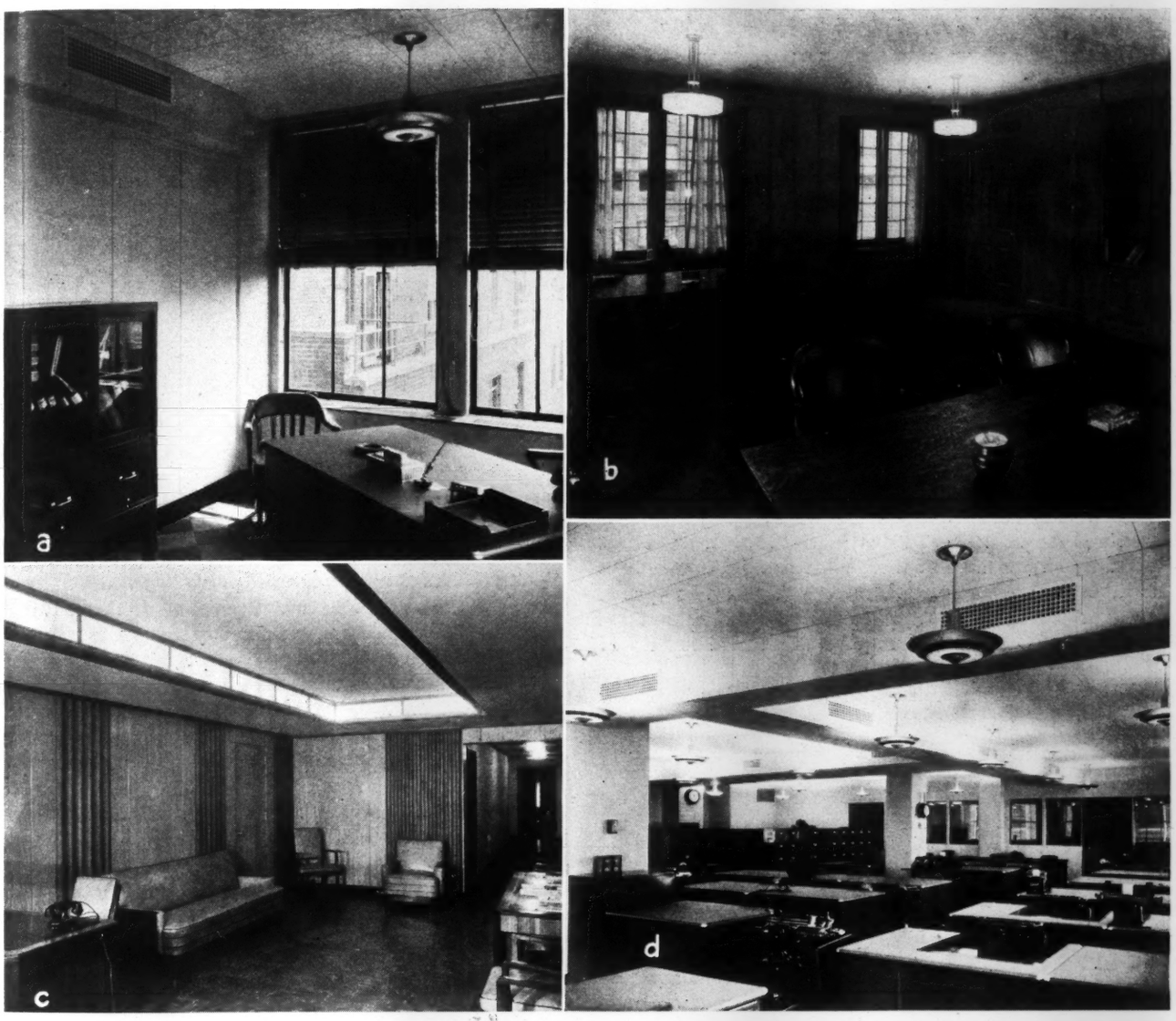


Fig. 4—(*a*) One of the many air-conditioned private offices; (*b*), an air-conditioned executive office; (*c*), the reception room for the executive offices; (*d*), air-conditioned clerical space. As is seen in these views the office layout is most modern; movable transite partitions are used as is ceiling acoustical treatment

circulating pump forces the condenser water from the washer tank through the condenser to the washer sprays. With minor changes in piping and air discharge connection, a second stage may be added to the present cooling tower to increase its capacity to meet future demands.

Automatic Control Electric Throughout

The automatic control is electric throughout—not only for the maintenance of conditions in the conditioned areas, but for the control features of the refrigerating equipment. Dew-point control of room conditions is effected from a thermostat in the dehumidifier-pump suction, controlling the mixture of cold and recirculated water to the sprays and controlling (in winter) the tempering coils and the mixture of fresh air and recirculated air to the dehumidifier. Dry-bulb control of room conditions is accomplished, first, through a thermostat in the return-air connection to each booster-fan room, controlling an automatic damper to regulate the amount of cold air supplied to booster-fan unit from the main supply-fan riser; second, from two room thermostats on each floor, one for inside rooms, one for outside rooms, controlling the reheater of each booster-fan unit; and the reheater mixing dampers. A velocity stat in the fresh-air intake on the 7th floor, acting as a limiting stat, insures a minimum volume of fresh air (not less than two air changes per hour), regardless of how the main supply fan distribution is throttled.

Condenser Water Requirements at Different Temperatures

The steam-jet refrigeration plant is designed to have an ultimate capacity to develop 250 tons of refrigeration effect, though at present only three thermal compressors have been installed, two each having a capacity of 70 tons and the third a capacity of 40 tons, a total of 180

(To page 68, back Advertising Section)

CHECKING UP on

Refrigeration Piping—

Suction and Discharge Lines for Ammonia

By **R. C. Doremus***

THE design of the refrigeration piping for a plant is governed not only by the over-all plant capacity, but more specifically by the size, capacity, and arrangement of the various individual units of equipment and apparatus comprising the whole plant. The size and location of the land on which the plant is located and its relation to lanes of traffic such as rivers, railroads, streets and highways often dictate the most desirable arrangement. Accordingly, no two plants (even of the same capacity) are exactly alike and the variables thus introduced require the fundamentals of good engineering design.

Whether an engineer is planning the piping of a new industrial refrigeration plant, an addition to an existing plant, or is checking up the piping of an old plant with a view towards its modernization, too much emphasis can hardly be laid on the importance of installing the proper size suction and discharge mains for handling ammonia vapor. Other ammonia main lines comprising the smaller miscellaneous piping connections must likewise be properly planned but these do not involve problems of velocities and pressure drop.

In many older plants many piping indiscretions are evident in the lay-out of the ammonia suction and discharge vapor lines.[1] If the pipe lines are not of ample size for conveying the required amount of vapor for the refrigeration tonnage involved, excessive velocities are bound to result which cause wire drawing, superheating and vibration of the piping. An abnormal pressure drop in a suction main line means that an ammonia compressor must operate at a lower suction pressure than would otherwise be required for the desired temperature in order to compensate for the pressure drop.

[1]For information on checking up refrigeration piping used for heat-transfer surface see page 194, April, 1933, issue; for data on brine and water piping, see page 294, June, 1933, issue.

***ADEQUATE** refrigeration piping will always pay for its capital investment; in fact, the plant will pay its cost in increased operating charges, less effective results, without it. Where too little thought was given the piping at the time of original installation, or where plant additions have changed the requirements, the piping must be brought up-to-date ..... This article tells how to assure correct, economical, ammonia suction and discharge piping.*

Inasmuch as the specific volumes of the vapor vary inversely with the pressure, lower suction pressures cause higher volumes of vapor to be handled per ton, as will be seen from the comparison given below; as the compressors are of a constant displacement, this naturally causes a decreased tonnage capacity from the machines. If more displacement is required of the ammonia compressor because of this, another compressor may be started if available, or, if only one compressor comprises the plant equipment it must run more hours per day to perform a given service. If the only compressor equipment is already running 24 hours per day, another machine must be purchased. Any of these alternatives is undesirable from a cost and efficiency standpoint. Accordingly, the tonnage capacity of a compressor may often be appreciably increased by increasing the size of small, over-burdened suction mains to lines of more adequate sizes.

Effects of Increasing the Suction Vapor Pressure

Regardless of the nature of the refrigerating work or the pressure at which that work is accomplished, it is generally desirable to run an ammonia compressor at the highest permissible suction pressure that will do the work. It is well-known that increasing the suction pressure of an ammonia compressor increases its tonnage capacity, but it also causes other interesting effects. To illustrate them, let us presume that an ammonia compressor is operating with a head pressure of 185 lb per sq in. gage and in one case it is operated with a suction pressure of 15 lb saturated and in the second case at a suction pressure of 25 lb saturated. Increasing the suction vapor pressure from 15 to 25 lb gage produces the following results:

1. Increases the heat content of the vapor from 611.6 to 615.4 Btu per lb of ammonia, or 0.62%
2. Increases the refrigerating effect from 460.7 to 464.5 Btu per lb of ammonia, or 0.82%
3. Increases the compressor tonnage capacity at constant speed by the inverse ratio of the displacement required per ton, or 6918 divided by 5243, or 31.7%
4. Increases the work done by the condenser due to the increase in tonnage capacity, or 21.7%
5. Decreases the weight of ammonia liquid circulated per ton of refrigeration from 0.4341 to 0.4306 lb per minute, or 0.8%
6. Decreases the specific volume of ammonia suction vapor from 9.236 to 7.047 cu ft per lb, or 23.8%
7. Decreases the displacement of vapor handled by the com-

*George B. Bright Co., Refrigerating Engineers and Architects, Detroit, Mich. Member of Board of Consulting and Contributing Editors.

pressor from 6918 to 5243 cu in. per minute per ton of refrigeration, or 24.3%

8. Decreases the temperature of the discharge vapor from 248 F to 221 F with adiabatic compression, or..... 10.9%

9. Decreases the indicated horsepower per ton of refrigeration from 1.240 to 1.028, or..... 17.0%

10. Decreases the heat content of the discharge vapor from 733.2 to 716.7 Btu per lb of ammonia, or..... 2.3%

Observations: There is a net increase in horsepower to drive the compressor due to the increase in tonnage capacity. For example, assume a compressor having a capacity of 100 tons of refrigeration at 15 lb gage suction pressure, then

$$\text{ihp} = 100 \times 1.240 = 124.0$$

Due to the increase in compressor tonnage capacity of 31.7 per cent, the capacity at 25 lb gage suction pressure is 131.7 tons, and

$$\text{ihp} = 131.7 \times 1.028 = 135.4$$

or a net increase in horsepower regardless of the mechanical efficiency of..... 9.2%

For each 1 lb of increase in suction pressure in this range, there is an increase afforded in ammonia compressor tonnage capacity of approximately 3 per cent and an increase in power of about 1.4 per cent, obviously advantageous.

Conversely, if the work to be done by the evaporator of the refrigerating plant may be done at 25 lb suction pressure and the piping is so inadequate as to cause a 10 lb friction drop (which is ridiculously large) the compressor would then have to operate at 15 lb suction pressure in order to overcome this friction drop. This would result in handicaps of the above-mentioned order and friction drops of less than this amount would be disadvantageous in proportion to their magnitude. This illustrates very strongly the vital necessity of matching the size of the suction main piping with the refrigeration capacity in order to reduce the friction pressure drop in the ammonia suction line to a practical minimum for economy in operation.

Effects of Increasing Discharge Vapor Pressure

Similarly, small discharge vapor lines with excessive line pressure drops mean that the compressor must labor under the disadvantage of a higher discharge pressure, and a higher mean effective pressure as well, than would be indicated by the condition of the condensing surface and the temperature of the condensing water. It is well-known that increasing the discharge pressure, with other conditions remaining constant, increases the power required to drive the compressor and decreases the compressor capacity but it also produces other interesting results. Presume that an ammonia compressor is operating at 15 lb per sq in. gage suction pressure saturated and that in one case the discharge pressure is 145 lb and in the other case it is operating at 185 lb per sq in. gage.

Increasing the discharge pressure from 145 to 185 lb per sq in. gage produces the following results:

1. Increases the weight of ammonia liquid per ton of refrigeration from 0.4196 to 0.4341 lb per minute, or..... 3.5%

2. Increases the displacement required of the ammonia compressor from 6696 to 6918 cu in. per minute per ton, or..... 3.2%

3. Increases the temperature of the discharge vapor with adiabatic compression from 213 F to 248 F. or..... 16.4%

4. Increases the heat content of the discharge vapor from 716.2 to 733.2 Btu per lb, or..... 2.4%

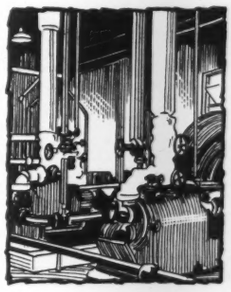
5. Increases the indicated horsepower per ton from 1.031 to 1.240, or..... 20.3%

6. Decreases the refrigerating effect from 476.7 to 460.7 Btu per lb of ammonia, or..... 3.4%

7. Decreases the compressor tonnage capacity at constant speed by the ratio of displacement—6696 cu in. per ton divided by 6918, or..... 3.2%

Observations: Although there is a decrease in compressor tonnage capacity, the mean effective pressure at which it is operating is higher. Consequently, the power required to drive the compressor is higher and naturally the horsepower per ton of refrigeration is higher inasmuch as fewer tons are being handled. If the ammonia compressor has a capacity of 100 tons of refrigeration at a discharge pressure of 145 lb per sq in. gage,

$$\text{ihp} = 100 \times 1.031 = 103.1$$



At a head pressure of 185 lb, the compressor capacity has been decreased from 100 to 96.8 tons but the ihp per ton has been increased from 1.031 to 1.240. In this case

$$\text{ihp} = 96.8 \times 1.240 = 120.0$$

showing an increase of 16.9 hp or 16.4%

The most significant results in this increase in discharge pressure are that for each 10 lb increase, there is a reduction in capacity of 0.75 per cent, an increase in ihp per ton of 5.0 per cent and a net increase in compressor horsepower of 4.4 hp or.....4% It is obviously most advantageous from a capacity and power-consumption standpoint to operate an ammonia compressor at a discharge pressure as low as practical. If the discharge lines are relatively short and of adequate size for the tonnage, the pressure drop will be a practical minimum and the discharge pressure will be determined by the condition of the condensing surface and the temperature of the condensing water available.

Practical Limits

Ammonia relief lines from ammonia pressure relief valves to diffusers or atmosphere should be installed the same size as the size of the outlet connection of the valve to avoid trouble with inspection bureaus. Such vapor lines as purge, pump-out and equalizer are seldom calculated from a velocity standpoint for then they would be ridiculously small. These are generally installed ¾ in. or larger and principally for structural strength of the pipe between pipe hangers.

In larger plants where oil drain lines are run from the various items of equipment to an oil purifier, these lines are usually no smaller than 1 in. to avoid grease fouling. This is quite essential where the oil is periodically withdrawn and purified and re-used along with new oil for economy.

Ammonia liquid feed lines are usually proportioned on the same basis as water lines although the liquid has only about 0.7 the weight of water. Here again, structural strength of the pipe between pipe hangers becomes a determining factor rather than fluid velocity.

The sizes of main piping for handling ammonia vapor must be chosen for suitable velocities and permissible pressure drops.

Some university research work has been done on ammonia-vapor velocities in straight lengths of pipe which will undoubtedly give very valuable information on this subject when released. Current practice of consulting and refrigerating engineers, based on past experience,

gives useful criteria from design or installation standpoints. The greatest value of such information is in connection with larger pipe sizes used in industrial plants of considerable tonnage capacity which would be difficult to reproduce in laboratory research set-ups. For this reason, existing plants give a wealth of data on vapor velocities that are safe to use from a practical viewpoint.

Maximum Recommended Vapor Velocities

In general, the refrigerating industry has been accustomed to short body sharp patterns in pipe fittings and long sweep fittings (similar to those used for water) are rarely found in plant piping systems inasmuch as they are not easily procured. Even though ammonia is of comparatively light density and might be handled at high velocities so far as pipe friction is concerned, the fittings are a limiting feature. No two plants are exact duplicates on account of varying layouts and capacities, yet the ammonia mains generally are made with a number of tees and elbow connections for branches as well as swing joints for flexibility and movement with temperature changes. Engineers who design and construct refrigeration plants have established criteria of limiting vapor velocities for economical operating costs. Their recommendations for maximum vapor velocities are indicated by Fig. 1, which will serve as a good guide for pipe and fittings used in common practice today.

Sizing the Suction Main

In special cases of suction pipe mains having fewer fittings than the customary plant or lines made of sweep fittings, or discharge lines that are unusually short, these velocities may safely be increased from 5 to 10 per cent without causing objectionable friction drops in pressure.

In order properly to design the ammonia suction and discharge mains, we are concerned with the amount of vapor to be conducted through these lines. This will vary with the tonnage capacity and the temperature, which defines the pressure. With variations in suction pressure, the specific volume of the suction vapor will vary greatly, thus requiring a displacement per ton of 1.37 cfm at 50 F, 3.92 cfm at 0 F to as great as 23.47 cfm at −63 F with 10 −F superheat. This is evidenced by the fact that a compressor used to handle 10 tons capacity for an ice-cream plant hardening room may need to displace four or five times as much vapor as a 10-ton machine would for manufacturing ice. Inasmuch as we know what these volumes are, the correct size of mains for any desired vapor velocity can easily be specified. Table 1 shows the ammonia suction vapor velocities at various suction pressures, each with 10 per cent superheat as would be the case with efficient operation, that would be caused by an amount of vapor representing 1 ton of refrigeration through various pipe sizes, from ½ to 10 in.

The following example indicates the use of the table in choosing desirable size of piping:

Let us presume we are considering an ice manufacturing plant operating at a suction pressure of 15.3 lb per sq in. gage and that we have a brine cooler of 70-ton capacity to handle. If one ton through a 3½-in. line at this pressure has a vapor velocity of 59 fpm, this 70-ton job will cause a vapor velocity of $70 \times 59 = 4130$. This size line is satisfactory inasmuch as it is just within the recommended limit of 4200 fpm. If we endeavored to use a 3-in. suction line from the cooler, the vapor velocity would be 79 fpm per ton, or $79 \times 70 = 5530$ fpm. This is too high and would cause an undesirable pressure drop due to friction in the mains and would require the compressor to work at a lower suction pres-

Suction Pressure Lbs.Gage	Corres. Sat. Temp. F	C.F.M. Per Ton	PIPE MAIN SIZES 1/2"	3/4"	1"	1¼"	1½"	2"	2½"	3"	3½"	4"	4½"	5"	6"	7"	8"	10"
19.7*	-63.1	23.47	14490	7828	4610	2630	1916	1010	713	462	346	268	214	169	117	87	68	43
17.7*	-57.6	18.97	11710	6322	3707	2123	1548	817	572	370	277	214	172	137	95	71	55	35
15.7*	-52.9	16.32	10070	5444	3190	1827	1311	702	492	318	238	184	148	117	81	61	47	30
13.6*	-48.6	14.42	8790	4808	2820	1614	1177	621	435	282	210	163	131	104	72	54	42	26
11.6*	-44.4	12.80	7903	4270	2504	1435	1045	551	386	250	187	144	116	92	64	48	37	23
9.6*	-41.3	11.97	7390	3992	2340	1340	977	515	361	234	175	135	108	86	60	44	34	22
7.5*	-38.1	10.56	6530	3520	2067	1183	862	454	318	206	154	119	96	77	52	40	30	19
5.5*	-35.2	9.72	6000	3242	1900	1088	793	418	293	190	142	110	88	70	48	36	28	18
3.5*	-32.4	9.02	5562	3005	1762	1010	736	388	272	176	132	103	82	65	45	34	26	17
1.4*	-29.8	8.34	5145	2782	1628	934	680	359	251	163	122	94	75	60	41	31	24	15
1.3	-25.0	7.47	4617	2490	1460	836	609	321	225	146	109	84	68	54	37	28	22	14
3.3	-20.6	6.57	4066	2190	1283	735	536	283	198	128	96	74	59	47	33	25	19	12
5.3	-16.6	5.96	3688	1986	1165	667	486	257	180	116	87	67	54	43	30	22	17	11
7.3	-13.0	5.42	3342	1805	1058	606	442	233	163	106	79	61	49	39	27	19	16	10
9.3	-9.6	5.00	3084	1666	977	560	408	215	151	98	73	56	45	36	25	18	14	9
11.3	-6.4	4.62	2850	1540	903	517	372	194	139	90	68	52	42	33	23	17	13	8
13.3	-3.4	4.30	2653	1433	841	482	350	185	130	84	63	48	39	31	21	15	12	8
15.3	-.6	4.03	2485	1344	788	452	328	173	122	79	59	45	36	29	20	14	12	7
17.3	2.1	3.78	2332	1260	739	423	308	163	114	74	55	43	34	27	19	14	11	7
19.3	4.7	3.55	2192	1183	694	397	289	153	107	69	52	40	32	25	18	13	10	7
21.3	7.1	3.36	2072	1120	657	376	274	145	101	66	49	38	30	24	17	13	10	6
23.3	9.4	3.20	1975	1067	626	358	261	138	96	62	47	36	29	23	16	12	9	6
25.3	11.7	3.04	1876	1014	594	340	248	131	91	60	44	34	27	22	15	11	9	6
27.3	13.8	2.90	1789	966	567	325	236	125	87	57	42	33	26	21	14	11	8	5
29.3	15.9	2.74	1690	913	536	307	223	118	82	54	40	31	25	20	13	10	8	5
31.3	17.9	2.65	1635	883	518	297	216	114	80	52	39	30	24	19	13	10	8	5
33.3	19.8	2.55	1574	849	498	285	208	110	77	50	37	29	23	18	12	9	7	5
35.3	21.7	2.45	1512	816	479	274	200	105	74	48	36	28	22	18	12	9	7	4
37.3	23.5	2.35	1450	783	460	263	192	101	71	46	34	27	21	17	11	9	7	4
39.3	25.2	2.27	1400	757	444	254	185	97	68	44	33	26	20	16	11	8	6	4
41.3	26.9	2.19	1352	730	428	245	179	94	66	43	32	25	20	16	11	8	6	4
43.3	28.6	2.12	1306	706	414	237	173	91	64	42	31	24	19	15	10	8	6	4
45.3	30.2	2.05	1266	684	401	229	167	88	62	40	30	23	19	15	10	8	6	4
47.3	31.8	1.97	1216	657	385	221	161	85	59	38	29	22	18	14	10	7	5	4
49.3	33.3	1.92	1185	640	374	215	156	83	58	37	28	22	18	14	10	7	5	3
51.3	34.8	1.87	1154	623	366	209	152	81	56	36	27	21	17	13	9	7	5	3
53.3	36.3	1.81	1117	603	354	203	148	78	55	35	26	20	16	13	9	7	5	3
55.3	37.7	1.77	1092	590	346	198	144	76	53	34	25	20	16	13	9	7	5	3
60.3	41.1	1.65	1018	550	323	185	135	71	50	32	24	19	15	12	8	6	5	3
65.3	44.4	1.55	957	517	303	174	126	67	47	30	23	18	14	11	8	6	5	3
70.3	47.3	1.46	901	486	284	163	119	63	44	28	21	16	13	11	7	6	4	3
75.3	50.5	1.37	845	457	268	153	112	59	41	27	20	15	12	10	7	5	4	3
Maximum velocity recommended			1600	2100	2500	2750	3000	3300	3600	4000	4200	4400	4600	4800	5000	5200	5400	6000

Table 1—Velocity of ammonia suction vapor in ft per min through various sizes of pipe lines for 1-ton refrigeration capacity at 185-lb gage condensing pressure

*Indicates vacuum in inches of mercury.

Note: Pipe sizes based on all suction lines 1½ in. and smaller being extra heavy, and all 2 in. and larger being full weight standard.

Velocities are based on vapor superheated 10 F above saturation. Each additional 10 F of superheat will increase velocity approximately 1 per cent. If discharge pressure is 150 lb gage, decrease velocities approximately 1 per cent.

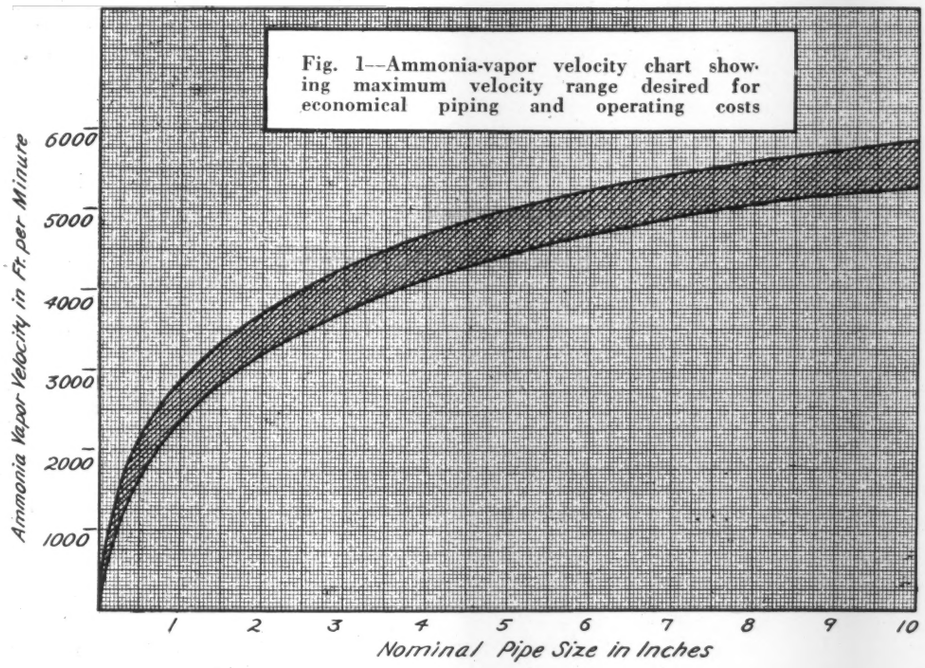


Fig. 1—Ammonia-vapor velocity chart showing maximum velocity range desired for economical piping and operating costs

sure to overcome this drop. This, of course, would be foolish economy inasmuch as very little would be saved by using a 3-in. line instead of a 3½-in. line, yet the power cost and longer hours of operation caused by this kind of scrimping would be excessive and would pay for the difference in initial pipe cost many times over. Where we would probably use a 3½-in line if the cooler were located fairly near the compressor, 4 in. would be preferable if the suction line were long, in order to favor pipe friction and also to use a size more easily procurable as a stock item.

Discharge Main Size

After the vapor has been compressed, its volume per ton has decreased considerably due to the compression, which is the reason the ammonia discharge line is usually smaller in size than the suction line. Table 2 shows ammonia discharge vapor velocities resulting from various suction pressures and indicates the velocity of vapor of 1 ton of refrigeration through various sizes of pipe lines in a similar manner to Table 1. This will similarly enable one to choose intelligently a suitable size of discharge line for any tonnage capacity without exceeding recommended limits. Of course, the length of discharge line may be very short, in which case these limiting values may be increased 5 to 10 per cent without any serious pressure drop.

Table 2—**Velocity of ammonia discharge vapor in ft per min through various sizes of pipe lines for 1-ton refrigeration capacity at 185-lb condensing pressure**

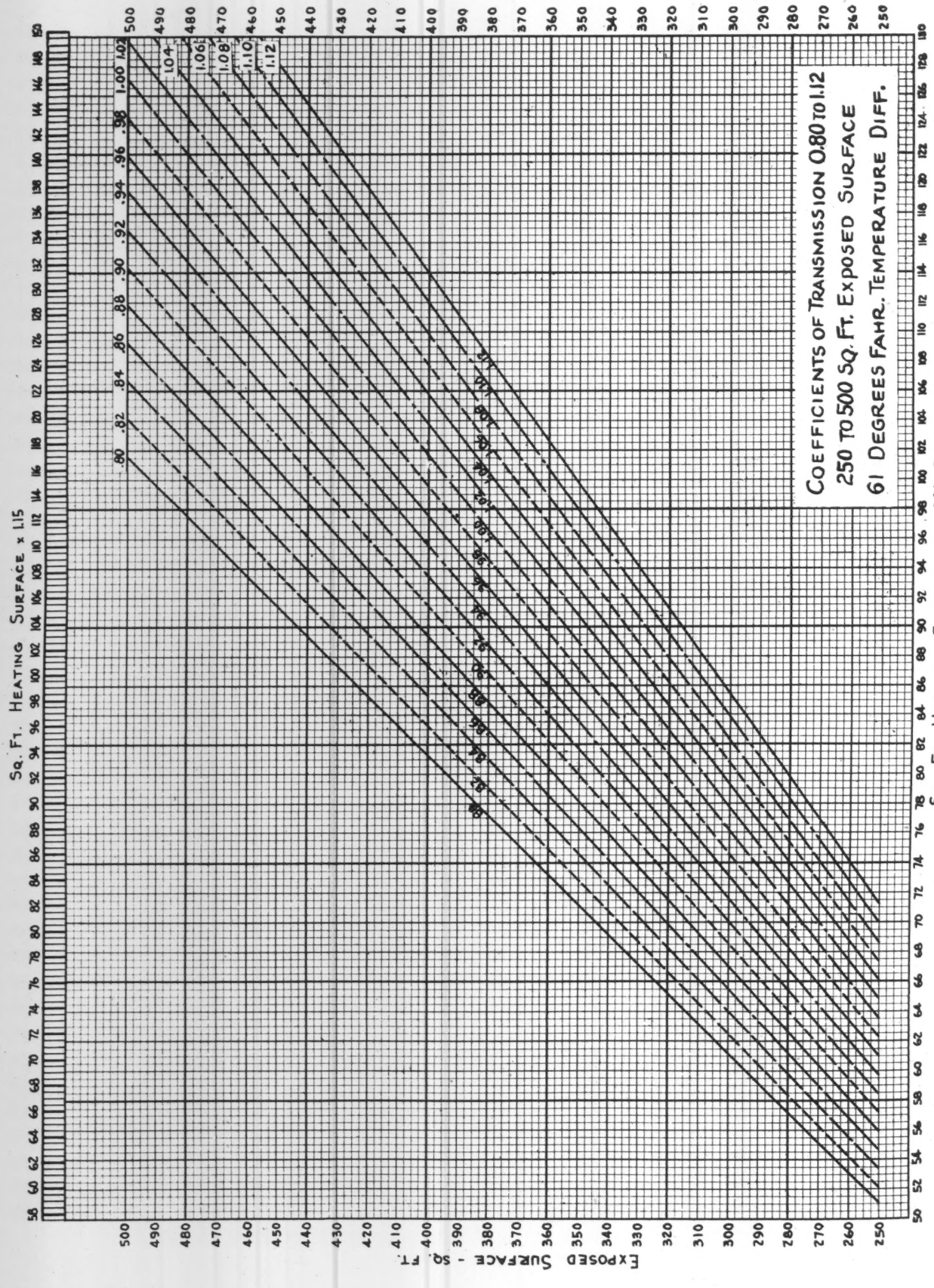
*Indicates vacuum in inches of mercury.

Note: Ammonia suction vapor is based on 10-F superheat before compression. Ammonia displacements per ton are based on adiabatic compression. All pipe sizes are extra heavy standard. If discharge line is extremely short, recommended maximum velocities may be increased 5 per cent.

Suction Pressure Lbs.Gage	C.F.M. per Ton	PIPE MAIN SIZES 1/2"	3/4"	1"	1¼"	1½"	2"	2½"	3"	3½"	4"	4½"	5"	6"	7"	8"	10"
19.7*	1.234	760	412	241	138	101	60	42	27	20	15	12	9	7	5	4	2
17.7*	1.218	750	407	238	137	100	59	42	27	20	15	12	9	7	5	4	2
15.7*	1.197	739	299	234	134	98	58	41	26	19	15	12	9	7	5	4	2
13.6*	1.175	725	392	230	132	96	57	40	26	19	15	12	9	7	5	4	2
11.6*	1.155	713	386	226	129	95	56	39	25	19	15	12	9	6	5	4	2
9.6*	1.127	694	376	221	126	94	55	38	25	18	14	11	9	6	5	4	2
7.5*	1.105	681	368	216	124	91	54	38	24	18	14	11	9	6	5	4	2
5.5*	1.087	670	363	212	122	89	53	37	24	18	14	11	9	6	5	3	2
3.5*	1.073	662	358	210	120	88	52	36	24	17	13	11	9	6	4	3	2
1.4*	1.057	652	352	206	118	87	51	36	23	17	13	11	8	6	4	3	2
1.3	1.034	638	345	202	116	85	50	35	23	17	13	10	8	6	4	3	2
3.3	1.010	622	337	197	113	83	49	34	22	16	13	10	8	6	4	3	2
5.3	.992	612	331	194	111	81	48	34	22	16	12	10	8	5	4	3	2
7.3	.968	597	323	189	108	79	47	33	21	16	12	10	8	5	4	3	2
9.3	.956	590	319	187	107	78	47	33	21	16	12	10	8	5	4	3	2
11.3	.944	583	315	184	106	77	46	32	21	15	12	9	7	5	4	3	2
13.3	.928	572	309	181	104	76	45	32	20	15	12	9	7	5	4	3	2
15.3	.917	567	306	179	103	75	45	31	20	15	11	9	7	5	4	3	2
17.3	.906	559	302	177	102	74	44	31	20	15	11	9	7	5	4	3	2
19.3	.894	552	298	175	100	73	43	30	19	14	11	9	7	5	4	3	2
21.3	.884	545	295	173	99	72	43	30	19	14	11	9	7	5	4	3	2
23.3	.875	540	292	171	98	71	42	30	19	14	11	9	7	5	4	3	2
25.3	.868	535	289	169	97	71	42	29	19	14	11	9	7	5	4	3	2
27.3	.860	530	287	168	96	70	42	29	18	14	11	9	7	5	4	3	2
29.3	.850	525	284	166	95	69	41	29	18	14	11	8	7	5	4	3	2
31.3	.843	520	281	165	94	69	41	29	18	14	11	8	7	5	4	3	2
33.3	.835	515	278	163	93	68	41	28	18	14	10	8	7	5	3	3	2
35.3	.828	511	276	162	93	68	40	28	18	13	10	8	7	5	3	3	2
37.3	.823	507	275	161	92	67	40	28	18	13	10	8	7	5	3	3	2
39.3	.817	504	273	160	92	67	40	28	18	13	10	8	6	5	3	3	2
41.3	.810	500	270	158	91	66	39	27	17	13	10	8	6	5	3	3	2
43.3	.803	496	268	157	90	65	39	27	17	13	10	8	6	4	3	3	2
45.3	.794	493	266	156	89	65	39	27	17	13	10	8	6	4	3	3	2
47.3	.792	489	264	155	89	65	38	27	17	13	10	8	6	4	3	3	2
49.3	.788	487	263	154	88	64	38	27	17	13	10	8	6	4	3	2	2
51.3	.786	485	262	154	88	64	38	27	17	13	10	8	6	4	3	2	2
53.3	.781	482	261	153	87	64	38	26	17	13	10	8	6	4	3	2	2
55.3	.776	478	259	152	87	63	38	26	17	13	10	8	6	4	3	2	1
60.3	.770	475	257	150	86	63	37	26	17	13	10	8	6	4	3	2	1
65.0	.756	466	252	148	85	62	37	26	16	12	10	8	6	4	3	2	1
70.0	.748	463	249	146	84	61	36	25	16	12	9	7	6	4	3	2	1
75.0	.743	458	248	145	83	60	36	25	16	12	9	7	6	4	3	2	1
Max. velocity recommended		1500	2100	2500	2750	3000	3300	3600	4000	4200	4400	4600	4800	5000	5200	5400	6000

Calculating Heating Surface for Buildings 6

. . . . by Price L. Rogers, Consulting Engineer, Philadelphia, Pa. This is Chart No. 6 of a series designed to enable computations for heat losses from buildings to be made accurately, with a minimum of time and with decreased possibility of errors. Chart No. 1 and an explanation of the series and its use was published in April; Chart No. 2, May; Chart No. 3, June; Chart No. 4, July; Chart No. 5, last month.


COEFFICIENTS OF TRANSMISSION 0.80 TO 1.12
250 TO 500 SQ. FT. EXPOSED SURFACE
61 DEGREES FAHR. TEMPERATURE DIFF.
Sq. Ft. Heating Surface x 1.15
Sq. Ft. Heating Surface at 240 B.T.U.
Exposed Surface - Sq. Ft.

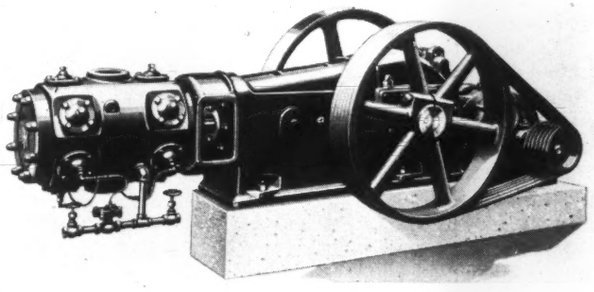
INTERESTING Equipment..... Developments

New Compressor Announced

A new single-stage, belt-driven compressor designed for heavy-duty service has been announced by Ingersoll-Rand Company, 11 Broadway, New York. It has one horizontal, double-acting cylinder and operates at moderate speeds. It is available in sizes from 10 to 125 hp, and for discharge pressures from 5 to 150 lb.

The machine is suitable where full-load, continuous service is required and will give economical standby service for large compressors whose full capacity is not always needed. It is adapted for use in isolated plants where there is little supervision, for applications where oil in the discharge line is objectionable, or for installations where a future change in pressure conditions may call for a change in cylinder size. It will handle poisonous or inflammable gases, or others that must be handled without leakage.

Low air speeds and pressure losses are obtained by liberal design of air passages and valves for economy. These features



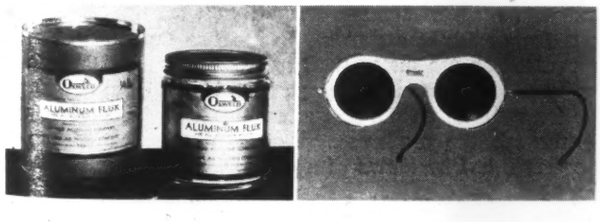
and effective water jacketing insure low air temperatures to simplify lubrication problems and lengthen the service life of valves, cylinders, and piston rings. A double row of Timken tapered roller bearings on each end of the crankshaft reduce friction. Regulating equipment suitable for any conditions of service can be furnished. A bulletin contains complete details.

Aluminum Welding Flux; New Goggles

A new all-purpose aluminum welding flux has been made available by The Linde Air Products Company, 30 E. 42nd Street, New York. It is intended to replace two fluxes previously marketed, one for welding pure aluminum and the other for welding aluminum alloys, and does everything that required the two separate fluxes to do. It is packed in ¼-lb jars which are packed in cardboard cartons.

This company has also announced new welding spectacles and a new lens. The lenses are mounted in a natural canvas bakelite frame and are 50 mm in diameter. This width permits a wide angle of vision and gives greater protection against light and sparks.

The temples (bows) are covered with insulating material, and the frame is non-flammable and does not conduct heat. By means of a snap device where the temples meet the frame it is possible to spread the frame and change lenses in a few seconds.



The new type lens, flat ground and polished, is made in light, medium and dark green shades. Use is recommended where safety codes prevent the use of other lenses which are not flat-ground and polished, although equally effective otherwise. This new lens conforms to all code requirements, including those of the Federal government.

New Automatic Timer Announced

A new electric timer that lends itself to many automatic and remote-control applications has been announced by the General Electric Company, Schenectady, N. Y. By combinations of two or more timers, or by using one in conjunction with other types of automatic time switches, it is possible to meet a wide range of process schedules.

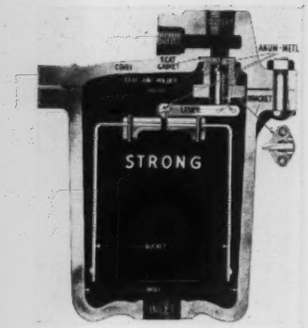
The timer is "all electric"—timing is started by closing a switch. Re-setting is automatic when the control circuit is de-energized. The timing period is readily adjustable over a wide range. When used as a process timer it can be arranged to operate a signal or terminate a process at the end of a pre-determined period. By the use of relays and timers together, entirely automatic control can be provided for almost any process.

There are two scales, graduated in hours or minutes according to the rating of the timer. One of these scales has a range three times that of the other. This permits a wide range of time-interval selection in a given timer. The user chooses the more appropriate scale for the interval he desires to time, by setting the scale selector lever. He then sets a movable contact at the point on the time scale corresponding to the interval to be timed. The timer is then ready for operation.

The timers are rated one ampere at 115 volts. For applications where the load to be handled exceeds the capacity of the contacts it is necessary to use a magnetic switch in conjunction with the timer. They are available for 60, 50 or 25 cycles, 115 or 230 volts, contacts normally opened or closed, and with time intervals of 5/15, 10/30, or 20/60 minutes.

Traps Have Several Features

A new series of inverted-bucket steam traps has been recently announced by the Strong, Carlisle & Hammond Company, Cleveland, Ohio. Features include the method of retaining the seat in place and the anti-balancing device.



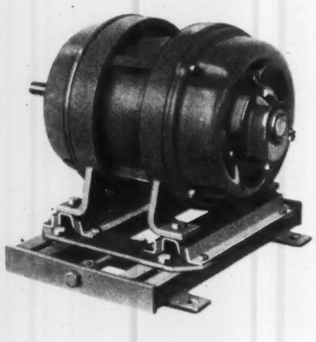
The seat is forced against a gasketed surface by a bracket which is held against the cover by two stud bolts. By simply removing the two nuts from the studs, the bracket and seat may be lifted out. This is aimed to eliminate the necessity of special tools and the possibility of damaging either the seat itself or the entire cover during replacement.

The patented anti-balancing device consists of a valve arrangement on the bucket which automatically varies the air-vent opening during the cycle of operation, which, the makers state, almost entirely eliminates the tendency to balance or dribble.

These traps are built of semi-steel construction for low and medium pressures and of forged steel construction for higher pressures and superheat service. A bulletin just issued describes both the semi-steel and the forged steel traps.

Sound-Proof Motor Base

A sound-proof motor base with an adjustment for belt tension has been developed by The Imperial Electric Company, Akron, Ohio, and is illustrated here. Incorporated in the construction of the base, supporting the weight of the motor and resisting the belt pull, are two lengths of silencing material. This material consists of supporting sections of brass-plated steel separated by live rubber which is vulcanized securely to the metal.

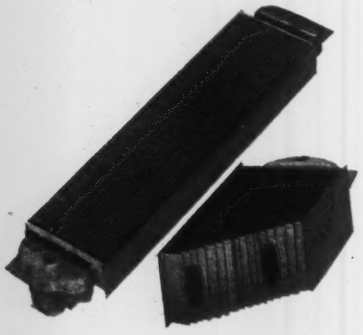


The arrangement is such that the rubber resists in shear the forces resulting from both motor weight and belt pull. This is a similar principle to the "floating power" for automobile engines. Under normal working conditions the stresses in the rubber-to-metal contact surfaces do not exceed 30 lb per sq in.

Installation of the base is not complicated by the sound-absorbing feature. Four anchor bolts are used as in the standard base; the only difference is that the motor is slightly higher.

Improves Convection Heaters

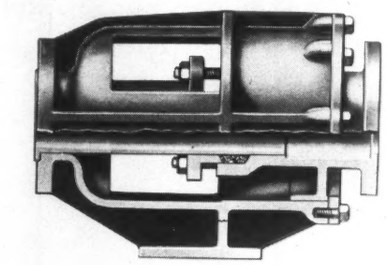
The Young Radiator Company, Racine, Wis., has recently improved the shape and design of tubes used in its convection heaters. The copper tubes are elliptical in shape and have been designed to conform more closely to the air flow through the core. Over 95 per cent of the tube is in continual contact with flowing air to prevent eddies. The tubes have large contact with fins and large bending strength. Large fin contact means increased conduction of heat to the fins and increased heating capacity. This so-called "streamaire" shape and lack of eddies increases air velocity through the core, according to the maker.

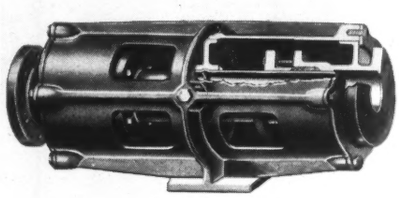


A patented brazing process is used in bonding the tubes and headers.

New Externally-Guided Expansion Joints

Two new externally-guided expansion joints have been developed by The Direct Separator Co., Inc., Syracuse, N. Y. In the single-slip type (upper view) the sliding member is composed of the stuffing box and guides, the external guide being enclosed in a cylinder extending the full length of joint, and the internal guide inside the stuffing box and adjacent pipe section. The external part is a single casting strong and heavily ribbed, carefully machined and concentric with all sliding parts. The joint can be installed as shipped since it is integral and the guides cannot be disturbed. The external guide is made relatively long and the distance from packing joint remains the same under all conditions. Ample openings are provided for repacking. It is made with or without bases which are machined when furnished. The lower view shows a double-slip joint of the same design with base. Joints are furnished in all sizes, standard and extra heavy, or of cast steel. There are no screw joints in the structure.





Air-Cooled Condensing Unit for Freon

A new air-cooled freon condensing unit, designed to meet a wide variety of circumstances where water-cooled condensers cannot be used, has been developed by the York Ice Machinery Corporation, York, Pa. It is made in capacities of 1, 1½ and 2 hp.

The unit is designed particularly to operate at evaporator temperatures up to 45 F, a suction temperature frequently required for air-conditioning work. It is especially adapted for use in circumstances where the cost of water is prohibitive; where the disposal of water is a difficult problem; where the water supply is never cold enough to do an effective job; and where the water is corrosive.

The new air condenser is mounted in front of the compressor and motor. All component parts are on a heavy cast semi-steel base with extended brackets integral with the base, which supports the condenser assembly. It has two fans, one of which is mounted on the motor shaft and draws the air through the condenser. The second is a specially-designed centrifugal suction fan mounted on the compressor flywheel, which also draws air through the condenser. Positive distribution of the air is accomplished by means of a sheet metal housing, which also

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Heating · Piping
and Air Conditioning

American Society of Heating and Ventilating Engineers JOURNAL SECTION

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Chart for Finding Mean Specific Heat of Air and Water Vapor

By **James W. May** † *(NON-MEMBER)*

Lexington, Ky.

THE accompanying chart shown in Fig. 1 was constructed for the purpose of having some accurate and expedient means of determining the mean specific heat of mixtures of air and water vapor.

The chart may be used for obtaining values of the mean specific heat in problems involving the heating of air from one condition to another, or in calculations pertaining to the cooling of air so long as the temperature of the air is not below the dew-point. It may also be used for finding values of the specific heat of mixtures of air and water vapor at any temperature or humidity within the range of the chart.

A portion of the Carrier Psychrometric Chart was used as a basis with the mean specific heat lines superimposed upon the same. These lines are based on the following formulae for specific heats:

For air, $C_p = 0.24112 + 0.000009T$
For water vapor, $C_p = 0.4423 + 0.00018T$
where T = temperature under consideration in degrees Fahrenheit.

The formula for air is known as Swann's Formula.

The formula for water vapor was taken from a paper presented to the *A. S. M. E.* by W. H. Carrier.

From the above formulae, C_p must be found independently for both air and water vapor, therefore this chart embodies a combination of the two and in the correct proportion depending upon the moisture content of the mixture. The relative humidity lines have been omitted. However, the moisture content for air at any condition may be found from the scale marked *Grains of Moisture Per Pound of Dry Air.*

In Fig. 2 is shown a small portion of the original chart drawn to a larger scale, with an example indicating the method to be followed in obtaining the mean specific heat.

Given the following problem:

What is the increase in sensible heat per pound of mixture of air at 71 F dry bulb, and 61 F wet bulb, when raised to a temperature of 129 F?

$H = C_p (T_2 - T_1)$
H = Sensible heat in Btu per pound.
T_2 = Final temperature = 129 F.
T_1 = Initial temperature = 71 F.

†Research Assistant, Heat Engineering Laboratories, College of Engineering, University of Kentucky.

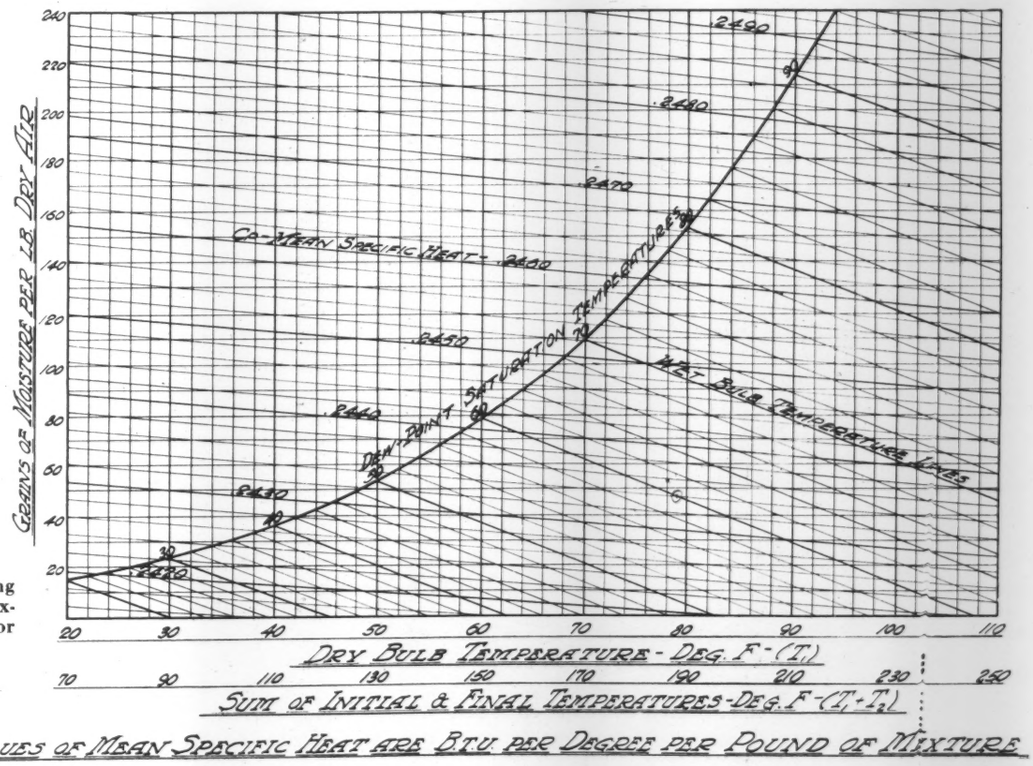


Fig. 1 — **Chart for finding mean specific heat of mixtures of air and water vapor**

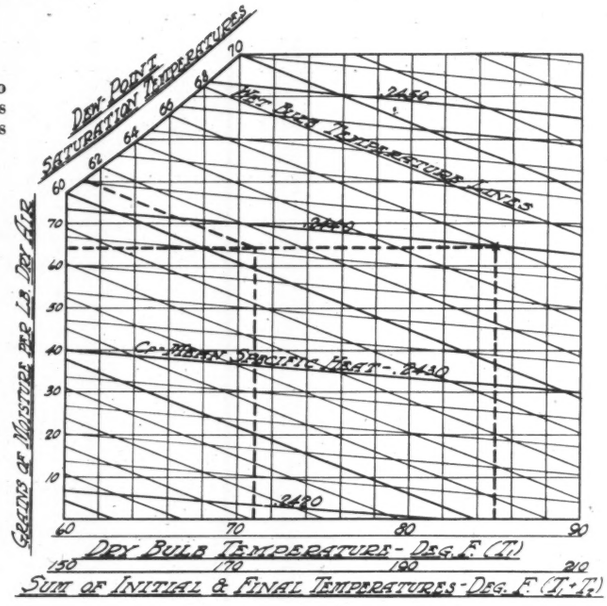

DEW POINT
SATURATION TEMPERATURES
WET BULB TEMPERATURE LINES
2440
2440
CO-MEAN SPECIFIC HEAT - 2440
2440
GRAINS OF MOISTURE PER LB. DRY AIR
DRY BULB TEMPERATURES - DEG. F. (T1)
SUM OF INITIAL & FINAL TEMPERATURES - DEG. F. (T1+T2)

Fig. 2—Example indicating method to be followed in obtaining the values of the mean specific heat of mixtures of air and water vapor

C_p = Mean specific heat of the mixture in Btu per degree per pound.

On the chart, Fig. 2, locate the intersection of the dry-bulb temperature line corresponding to 71 F, and the wet-bulb line corresponding to 61 F. This gives the value of the grains of moisture per pound of dry air. Follow this horizontal line indicating the grains of moisture per pound of dry air until it intersects the vertical line corresponding to 200 F which represents the sum of the initial and final temperatures $(T_1 + T_2)$. At this point read $C_p = 0.2440$, the value of the mean specific heat.

Then $H = 0.2440\ (129 - 71)$
$= 14.15$ Btu increase in sensible heat per pound of mixture.

Cleveland Chapter Elects Officers

May 16, 1933. The annual meeting was called to order by Pres. R. G. Davis at 8:00 p. m. in the Chapter Rooms of the *Cleveland Engineering Society* with 37 members and guests in attendance.

Following the reading and approval of the minutes of the April meeting, Mr. Kilstadt of the *Cleveland Engineering Society* explained a plan for attending Engineers Week at the Century of Progress in Chicago.

W. E. Stark of the Membership Committee reported that, through the activities of R. H. Wehrle, a student chapter was being organized at Case School.

The Auditing Committee, composed of E. J. Vermere and C. A. Wheeler, reported that the books and accounts had been checked and found to be correct.

Secy. F. A. Kitchen told of the number of meetings held during the year and the status of membership dues.

The activities of the Unemployment Bureau of the *Cleveland Engineering Society* were discussed and it was suggested that a written notice be sent to the members of the Chapter so that they would be familiar with these facilities. On motion of Mr. Stark, seconded by Mr. Vermere, a contribution of $10.00 was voted towards maintaining the Unemployment Bureau.

The report of the tellers of election showed that in view of the ballots cast the following officers were elected for the ensuing year:

President—F. A. Kitchen
Vice-President—M. F. Rather
Secretary—R. H. Wehrle
Treasurer—H. M. Nobis
Board of Governors—R. G. Davis, Philip Cohen and J. W. Farley

J. C. Miles, Program Committee Chairman, introduced the speaker of the evening, C. A. Burton, whose subject was Welded Steel Frame House Construction and Application of Heating and Air Conditioning. Mr. Burton discussed the progress made in this method of residential building and showed slides and motion pictures taken during the construction of steel houses in the Cleveland district.

Mr. Burton then introduced C. C. Peck, contract welder, who had a complete electric welding demonstration unit set up in the meeting rooms. His representative then demonstrated the process of welding members, beams and similar work required in building a steel frame house.

This unusual exhibit was exceedingly interesting and instructive and a rising vote of thanks was extended to those in charge of the meeting for the program presented.

Current Literature on School Buildings

A Bibliography of School Buildings, Grounds and Equipment in three parts has been compiled under the direction of Henry Lester Smith, Director of Bureau of Cooperative Research, School of Education, Indiana University, Bloomington, Ind.

Volume IX, No. 3 (Part III) was issued in June, 1933, and continues the consecutive numbering of references and covers the period up to April 1, 1932.

The subjects listed under Mechanical Equipment are: Interior Decoration, Standardization of School Buildings and Equipment, Remodeling, Maintenance of Buildings and Equipment, Insurance and Depreciation, Teacher's Home, Open Air Schools, Portable Schools, Schoolhouses of Other Countries, Plans, Illustrations and Descriptions of Various Types of School Buildings.

In the list relating to Mechanical Equipment of buildings, which includes heating and ventilation, sanitation, water supply, lighting, fire protection and cleaning equipment, generous reference has been made to the Society's publications as well as to other sources in the fields covered.

These publications may be obtained through the University of Indiana at a cost of fifty cents per copy.

John W. Waters Dies

John W. Waters of Johnstown, Pa., president and a founder of the National Radiator Corporation and president of the United States National Bank at Johnstown died in Emergency Hospital, Washington, D. C., August 14, of a cerebral hemorrhage he suffered while attending a banking conference at the Treasury last Thursday. He was seventy-three years old.

Mr. Waters was stricken when he arose to reply to Mayor Edward McCloskey of Johnstown, who had opposed the reorganization plan just offered by Mr. Waters for reopening the Johnstown bank, of which he was head.

The National Radiator Corporation, with which Mr. Waters had been associated most of his life is one of the largest manufacturers of heating equipment in the country with main offices at Johnstown.

Mr. Waters, the son of a Civil War veteran, was a member of the Masons, Knights Templar, the Elks, the Manufacturers' Club of Philadelphia and several Johnstown civic and social clubs. His wife and three children survive.

Temperature Gradient Observations in a Large Heated Space*

By G. L. Larson†, D. W. Nelson‡ *(MEMBERS)* and O. C. Cromer§ *(NON-MEMBER)*

Madison, Wis.

This paper is the result of research conducted by the Mechanical Engineering Department at the University of Wisconsin in co-operation with the American Society of Heating and Ventilating Engineers' *Research Laboratory.*

RESEARCH data concerning stratification of air in buildings are lacking and, when confronted with an unusual type of building, the designer often finds himself literally at sea in selecting inside temperatures upon which to base heat loss calculations.

According to The A. S. H. V. E. Guide, it is the common practice of engineers to allow 2 per cent per foot of height above the breathing line in establishing the probable air temperatures at any given level for a direct radiation system, and not less than 1 per cent per foot above the breathing line in arriving at the probable air temperatures at any given level when a combined heating and ventilating system is used.

Fig. 1—View of field house at the University of Wisconsin

While values thus obtained are sufficiently accurate and undoubtedly safe in most cases, this method of calculation leads to erroneous and even absurd results when applied to large open rooms with unusually high roofs or ceilings. It was for the purpose of obtaining data on this subject that this research was undertaken.

The present study of temperature gradients has been carried on in the University of Wisconsin Field House, a building which is used primarily for basketball games and for indoor track. Fig. 1 shows the exterior of the building. The interior dimensions of the main part are as follows: width 193 ft, length 227 ft, height 97 ft. In addition to the main space, there is an additional space formed under the bleachers of the adjacent football stadium. The ⅛ mile running track goes through this space and the fan rooms, locker and shower rooms are also located here. The total internal volume of the building is about 3,500,000 cu ft.

The plans for the building provide for an ultimate seating capacity for basketball purposes of about 11,500. This capacity is to be secured by the use of two balconies accommodating 8,000 and wooden bleachers seating 3,500. The upper balcony has not yet been built. With temporary bleachers built up from the playing floor, the largest capacity secured up to the present is about 6,000. Fig. 2 shows a plan and elevation of the building. The lower balcony extends from the 13 ft elevation to the 25 ft elevation. The second balcony will occupy an elevation from 29 ft to 50 ft. Fig. 3 shows the general relation of the balconies and the playing floor. The support members for the upper balcony show in this view. The playing floor is large enough for two practice courts cross-wise of the building. For competitive games, the court lengthwise of the building is used as shown in the photograph. Bleachers are erected on all sides of the playing floor for extra seating capacity. The floor is made up of bolted sections which are dismantled and stored at the end of the basketball season. The support is on wood posts which rest on concrete blocks a few inches under the surface of the dirt floor. The dirt floor is used for spring football and baseball practice as well as for indoor track meets.

The exterior walls are of Madison sandstone backed up with poured concrete. The building is well lighted by large windows. The roof construction is 2 in. plank, 1 in. of cork and tile. Double-glazed sky-lights are located in the center of the roof as shown in Fig. 1.

*Presented at the Semi-Annual Meeting of the American Society of Heating and Ventilating Engineers, Hotel Statler, Detroit, Mich., June, 1933.

†Professor of Steam and Gas Engineering, University of Wisconsin.
‡Assistant Professor of Steam and Gas Engineering, University of Wisconsin.
§Instructor in Steam and Gas Engineering, University of Wisconsin.

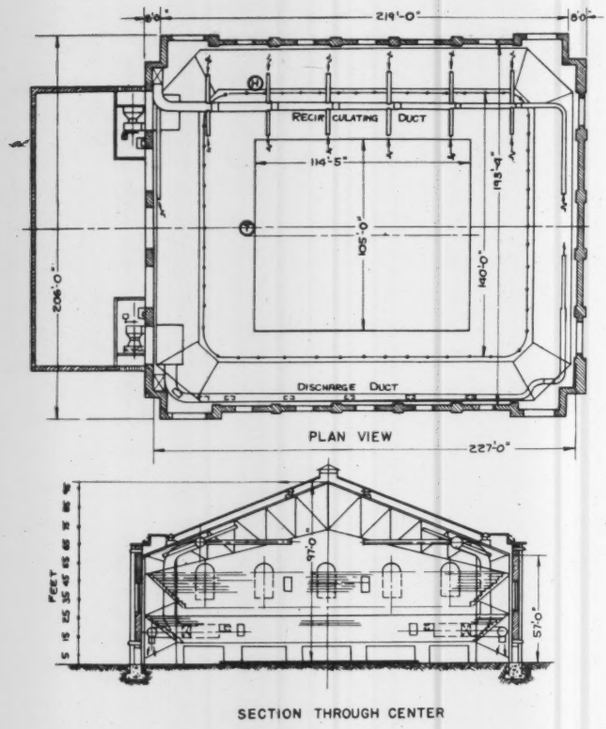


Fig. 2—Plan and section of the field house showing the recirculating and discharge ducts

Description of Heating and Ventilating System

During games, it is desired to keep the playing floor relatively cool. The spectators require a somewhat higher temperature. Since they are located in balconies at a higher elevation, the natural tendency for heated air to rise, aids this desired differential heating. However, the spread of the balconies over an elevation from 13 ft to 50 ft makes the problem of securing uniform temperature for all spectators serious. During periods of inoccupancy, it is desired to keep temperatures at all elevations low for the sake of economy.

In order to keep roof temperatures within reasonable limits and to fulfill the desired conditions at the balconies and playing floor, a fan system of heating was decided upon. Since the building is used only a few hours at a time and the air volume per person is large, ventilation at all times of occupancy is not required. At times the moisture content may be high due to evaporation from the dirt floor when sprinkled, in addition to that given off by the occupants. This makes it desirable to have available an outside air supply. Further, in mild weather, the heat given off by the occupants may need to be dissipated by supplying outside air.

The heating and ventilating system is in two duplicate parts and is shown in Fig. 2 in plan and elevation. The two fans have each a measured capacity of 55,000 cfm. The air supply may be taken from outside entirely or the system may recirculate a certain proportion or all of the air. The resistance of the outside air supply system is equal to the resistance of the recirculating duct system. The recirculating inlets are located near the roof. For each fan, there are six 30 in. inlets located 58 ft from the floor, six 24 in. inlets at an elevation of 73 ft and three 30 in. inlets at each end of the building at an elevation of 66 ft.

Three tempering coils each consisting of 1050 lineal feet of ⅝ in. finned copper tubes two rows deep are located side by side in each fan chamber. The air discharge is from a trunk line located under the lower balcony. There are twelve 29 in. by 56½ in. outlets directed downward in each of the two discharge ducts. Each outlet is equipped with a heater consisting of 212 lineal feet of ⅝ in. finned copper tubes placed three rows deep. Each one of these outlet heaters is under control of a thermostat placed on the structural columns above the lower balcony at an elevation of about 25 ft. These thermostats are set to maintain the temperature desired in the balconies. The action of the system when recirculating is to draw the warm air from the upper levels near the roof, heat it to the desired temperature at the various outlets and allow it to rise up through the open spaces of the balcony.

When arranged to take an outside supply, the air is heated to about 55 F in the tempering coils and is then heated to the desired balcony temperature at each individual outlet. Twenty 36 in. diameter roof ventilators are provided in four rows, each row having an independent manual damper control.

No direct radiation is used except in the shower, toilet and locker rooms which total 1072 sq ft. The supply and return mains of the vacuum steam system are not covered and constitute direct radiating surface. The area of steam supply mains and branches is 1,524 sq ft. The lights over the playing floor also have considerable capacity as heating elements. Over the playing floor,

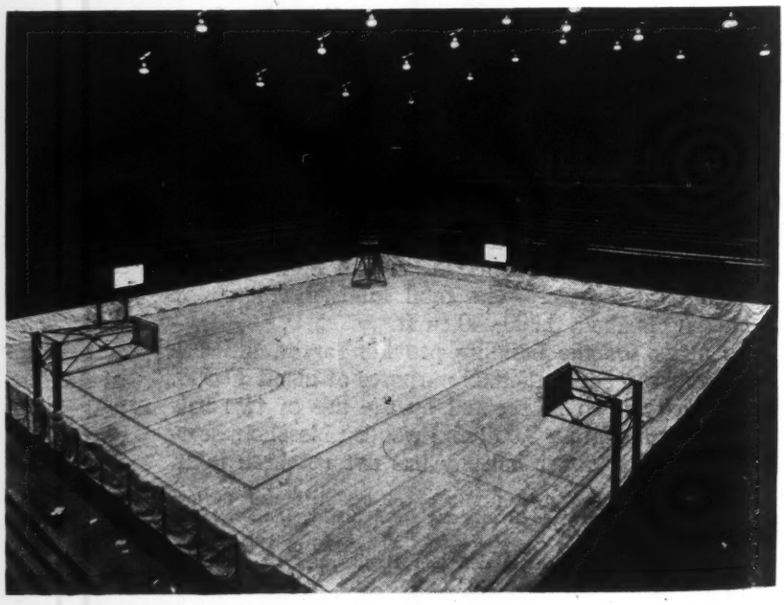


Fig. 3—Interior view, showing playing floor

there are thirty 1500 watt lights and twelve 1000 watt lights totalling 57,000 watts. These are at an elevation of 58.5 ft to 73 ft.

Nature of Observations

Observations under various conditions as to occupancy, weather and operation of system have been made during the three seasons the building has been in use. In order to obtain the temperatures from floor to roof, ten resistance thermometers were hung from the peak of the roof structure at one end of the playing floor. The first thermometer was placed at the 5 ft level and the last one at the 95 ft level with 10 ft spacing between them. Readings were taken by means of a Wheatstone bridge arrangement through a 10-point switch. The thermometers were checked against mercury thermometers by immersion in a water bath. Approximately 1000 ft of duplex lighting cord was used to make the circuits. The location of the thermometer cable is shown on the plan view marked as *T* on Fig. 2. Relative humidities were determined directly by a recording instrument. The wet and dry bulbs for this instrument were located at the point marked *H* on Fig. 2 on a beam directly below the lower part of the balcony in line with the resistance thermometer cable. Checking of temperatures showed that the values here generally corresponded closely to the temperatures obtained at the same elevation of 15 ft in the center of the building.

The steam used was measured by a recording steam flow meter placed in the main supplying the building. The amount recorded included a certain amount of pipe line loss and also the steam used to heat the shower water.

Observations have been taken during all of the basketball games of the past three years. In addition, readings have been taken during times of non-occupancy such as on zero days to observe gradients during still air conditions. The results of these various observations are shown in Figs. 4 to 9. The conditions of operation were established by the janitor, who decided when to operate the fans recirculating or with outside air and whether some or all of the roof vents should be open. In general during the first two seasons a temperature of 55 F was desired at the playing floor. During the past season, this desired temperature has been increased to 60 F or slightly above. The balcony thermostats have been set for 65 F during all three heating seasons.

Methods of Operation of System

There are several methods of operation of the system and each has been used during some of the games. Fig. 4 shows results of operation in mild weather and with a large attendance without the fans running, that is, with gravity operation. Fig. 5 presents the observations using recirculation in fairly cool weather. Fig. 6 shows results with the use of outside air and partial recirculation during a game with a small attendance in near zero weather. During this game, all outside air was used for a period. Fig. 7 shows similar operation in mild weather and with a larger attendance. During all of these games, the roof vents were open.

In addition to the results of observations during occupancy shown in Figs. 4 to 7, results obtained during cold weather and no occupancy are shown in Figs. 8 and 9. The main purpose of these latter figures is to show the temperature gradient under severe heating conditions and with no disturbance of stratification tendencies. In Fig. 8 the influence of the starting of the fan on the still air gradient is studied. In Fig. 9 the gradients observed on three below-zero days are shown.

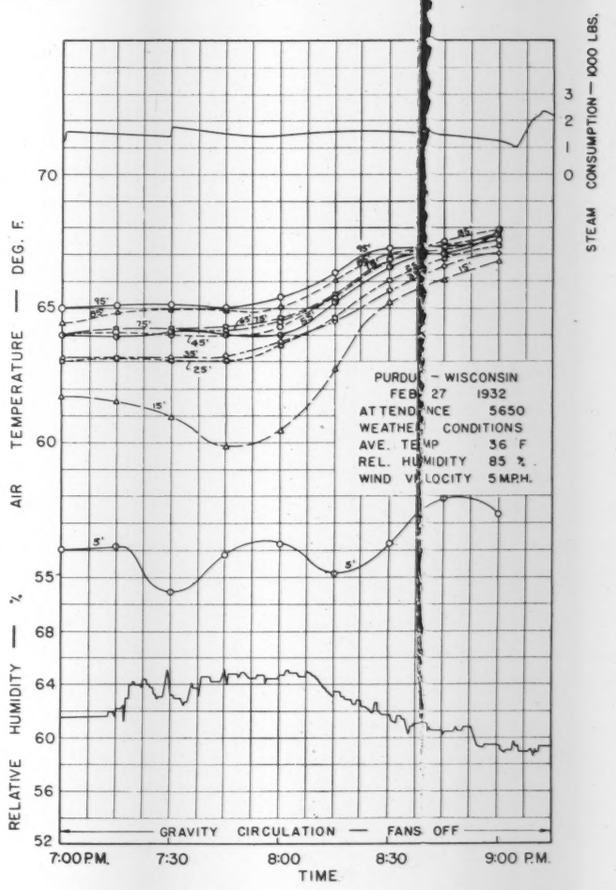


Fig. 4—Record of temperatures, humidities, and steam consumption with gravity circulation—fans not running

Gravity Circulation

During gravity operation of the system, the circulation through the ducts is in the opposite direction to that obtained with fan recirculation. The rate of air circulation depends upon the extent of the heating of the air as it travels upward through the ducts. The air enters at the discharge outlets below the lower balcony and is further heated by the outlet heaters if the balcony thermostats are calling for heat. Since the capacity of the system is very much limited under gravity circulation, under all but mild weather conditions steam would be present in these outlet coils. In mild weather only one of the three tempering coils is kept on. The air then, on its upward travel receives but little heat in the fan rooms since its flow is divided between the area consisting of the by-pass and the two tempering coils not supplied with steam, and the area of the one tempering coil which is supplied with steam.

Fig. 4 shows the results secured with such circulation

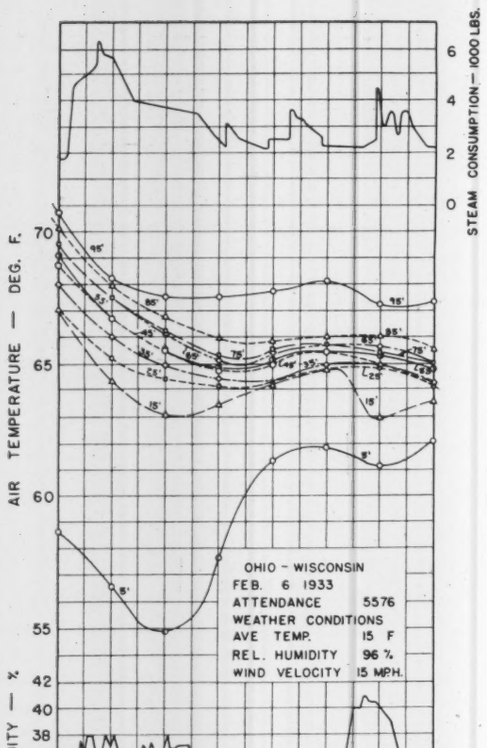


Fig. 5—Record of temperatures, humidities, and steam consumption using totally-recirculated air

when the outside temperature was 36 F and when a capacity crowd of 5650 was present. These conditions would seem to indicate that outside air might well be introduced by fan operation for cooling. The heating capacity of the system standing idle was sufficient to bring the average temperature of the balcony elevations to about 63 F as shown at 7:00 p. m. The air immediately above the playing floor was about 56 F in temperature which was satisfactory. The rise in playing floor temperatures from 7:30 to 8:00 p. m. indicated that outside air was needed for cooling. Some windows were opened at about 8:00 p. m. which helped to keep the rise at the playing floor to a total of 3 F. The rise at the 15 ft level was 7 F. The rise at the balcony levels which are from 13 ft to 50 ft was about 4.0 F. The variation in temperature from highest to lowest point of the balconies was 3.8 F at 8:00 p. m. and decreased to 1.0 F at 9:00 p. m. This small difference of 1.0 F appears to be due to the uniform emission of low temperature heat by the large audience. Since the balcony thermostats are set at 65 F, the outlet heaters probably shut off sometime after 8:00 p. m. The highest temperatures are not at the roof but about mid-way from floor to roof at 9:00 p. m. The lights over the playing floor are at this elevation and may cause this higher temperature. The heat equivalent of 57,000 watts is sufficient to raise the volume of air above the lights about 19 F in one hour or is sufficient to raise the entire volume of 3,500,000 cu ft 3.1 F. This is almost equal to the rise observed from 8:00 to 9:00 p. m.

The audience and players have a high capacity as heaters. Assuming the average individual present is emitting 350 Btu of sensible heat per hour, this would be a total of 1,980,000 Btu. This is available for heat transmission losses and to heat up air passing in and out of the building. The heat transmission losses are estimated as 965,000 Btu at the average outside tem-

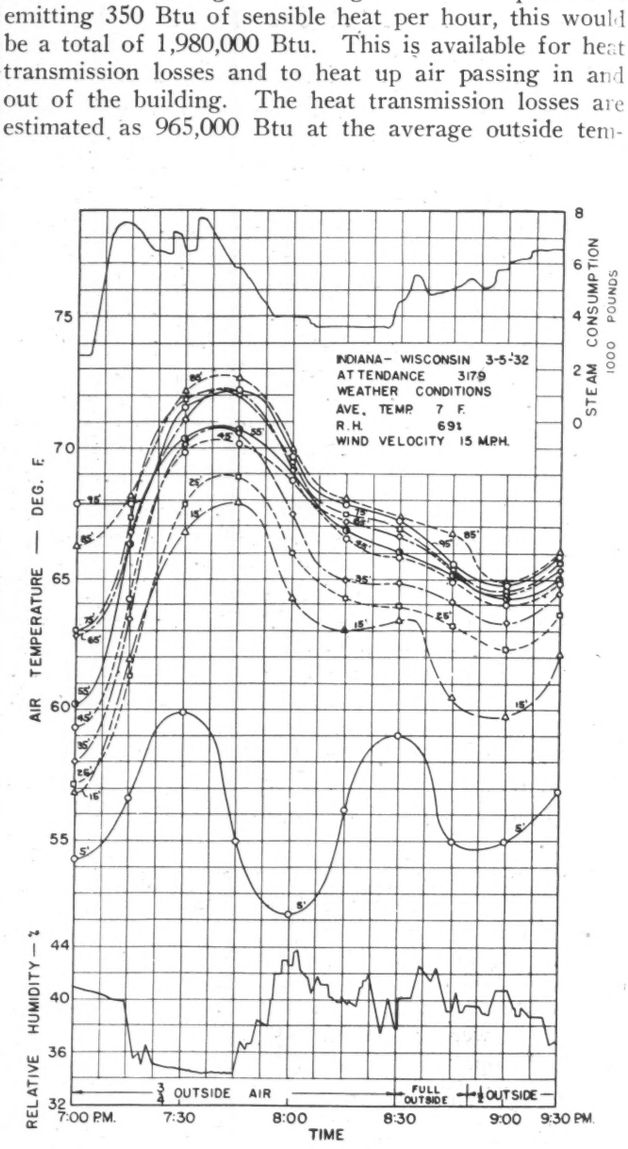


Fig. 6—Record of temperatures, humidities, and steam consumption using outside air in cold weather

perature of 36 F prevailing during this game. The almost 1,000,000 Btu of heat thus available will warm up about 1,736,000 cu ft of outside air to inside temperature which amounts to one-half air change per hour. The amount of steam used for space heating is not known because some of the supply is used for heating hot water and because a pipe line loss is also included. Assuming it is 1000 lb of the 1500 average total, another one-half air change per hour would be required to dissipate this heat. Based on one air change the velocity through the twenty 36 in. roof vents would be 410 fpm.

The absolute humidity increased 0.46 grain per cubic foot from 8:00 to 9:00 p. m. assuming that the dry bulb temperature at the humidity pick-up element un-

der the balcony was the same as that at the 15 ft level in the center of the floor. On several occasions this has been checked and found to be very nearly correct. This increase in moisture was due to that given off by the occupants. During this period the air entered with a humidity of 2.1 gr and left with 3.91 gr at the beginning and with 4.37 gr at the end of the period. The 5650 people would be adding 808 lb of moisture per hour to the air, based on an amount of 1000 grains being given off per hour by each person. The following equation balances the moisture added by the audience against the moisture increase in the room volume and that carried away in the ventilating air.

$$808 = \frac{3{,}500{,}000 \times 0.46 + 2.04 \text{ (cfh)}}{7{,}000}$$

This indicates an air movement of 1,983,000 cfh which is an air change of 0.57. Any moisture evaporated from the dirt floor which is sprinkled at times would indicate a greater air change than the above 0.57. If the air change is one per hour, the moisture evaporated would be 442 lb.

At no time during any game has fogging appeared over the playing floor. Conditions seemed not far from causing this at 7:30 p. m. when the dry bulb was 54.5 F and the dew point 49 F. This is a good reason for using ventilation. At times some condensation has occurred on the roof ventilators and on the skylights. The roof insulation has prevented condensation on its under surface under all conditions encountered.

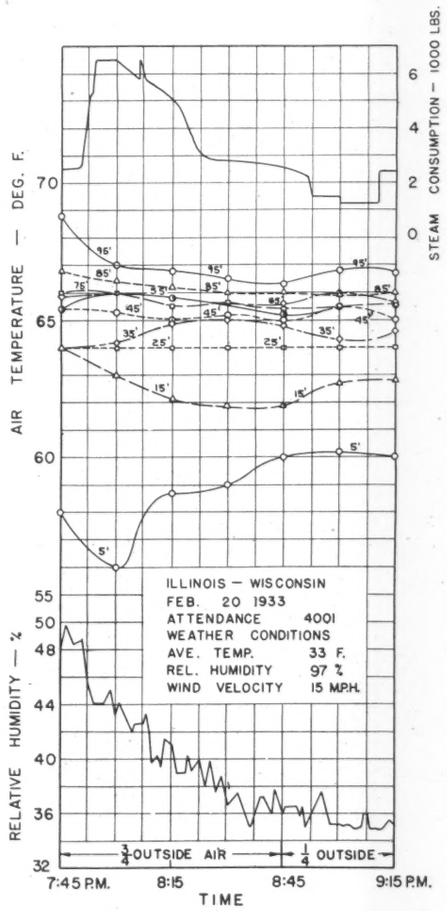


Fig. 7—Record of temperatures, humidities, and steam consumption using outside air in mild weather

System Operated Recirculating

Fig. 5 shows the operation of the fan system recirculating the building air during the first one hour and twenty minutes of observation. From 7:30 to 8:00 p. m., a general drop in temperatures is shown due to the doors being open as the crowd arrived. Heat is being added but at an insufficient rate to maintain temperatures. At the start of this period, there seems little doubt but that the balcony thermostats were above 65 F so that the outlet heaters were closed. These probably came on intermittently during this half hour period due to cool air currents from the open doors. The steam consumption curve shows a considerable peak during this period.

From 8:00 to 8:50 p. m., the temperature above the 15 ft level changed but little with a rise of about one-half degree. The rise 5 ft above the playing floor was from 55 F to 62 F during the same period. Since the audience is largely above this level, the rise probably is not due to this source of heat. It seems due to discharging from the outlets below the balconies air that entered the recirculating ducts at about 65 F and which received little heat from either the tempering coils or the outlet heater coils. The by-pass around the tempering coils would be open since the temperature of the recirculated air is about 55 F. The air was discharged then at somewhere near 65 F and since it was not much warmer than the air above the playing floor, it diffused at these lower levels. Had it been considerably warmer, its course would have been up through the balconies. The result was a 7 deg rise 5 ft above the playing floor in 50 min. A rise of 2 deg to 65 F was observed at the 15 ft level. This indicates a probable discharge temperature of 65 F.

The introduction of some outside air from 8:50 to 9:05 p. m. resulted in lowering the discharge temperature. The resulting drop in temperature at the 15 ft level was 2 F and at the 5 ft level was less than 1 F. This indicates a probable discharge temperature of 60 F to 61 F. The change back to entire recirculation at 9:05 p. m. resulted in the same rising temperature at the lower levels as experienced from 8:00 to 8:50 p. m. The average temperatures at balcony levels were 64.4 F, 64.7 F, and 64.2 F at 8:00, 8:30, and 9:00 p. m. respectively. The corresponding temperature variations were 2.5 F, 1.0 F, and 2.4 F.

The humidity record shows no definite change. From 8:00 to 8:50 p. m. with recirculation, the relative humidity decreased while the 15 ft level temperature increased correspondingly. The absolute humidity shows no significant change, although an increase would be expected from the large audience.

Fan Operation with Outside Air

Fig. 6 shows the introduction of outside air with an outside temperature of 7 F average. The attendance was about 3200. At 7:00 p. m., the temperature varied from 54.3 F at the 5 ft level to 67.8 F at the 95 ft level. The temperature at the balcony thermostat level was 57 F which shows the heating system under still air con-

ditions was insufficient to bring the temperature up to the 65 F established for the thermostats. The starting of the fans with ¾ outside and ¼ recirculated air caused the steam consumption to increase from 2,500 to 7,500 lb per hour. Under these conditions, the air at all levels increased rapidly in temperature. This increase was about 5 F in one half hour. At 7:20 p. m., 65 F was reached at the 25 ft level. This would indicate that the thermostats would prevent further heating at the outlet heaters. However, the steam consumption did not start a definite decline until 7:35 p. m. when the temperature at the 25 ft level was 68 F.

The temperatures directly above the playing floor started to decline at 7:30 p. m. and continued until 8:00 p. m. when the game started. This drop of 8 F is considered due to the opening of doors while the crowd entered. From 8:00 to 8:30 p. m., a rise of 7 F was observed at the 5 ft level while a general decline was noted at higher levels. It seems likely that the temperature at balcony thermostats was such as to cause the steam to be shut off from the outlet heaters. If so, the discharge temperature depends upon the resultant mixing of the outside and recirculated air. With the 8 F outside temperature and with temperatures of 67 F at the recirculating inlets, the mixture temperature would be about 37 F. This would cause the by-pass to be closed and the tempering coils would raise this air to 55 F. The low steam consumption of 3600 lb per hour during this period and the closeness of the temperature at the 25 ft level to 65 F would seem to indicate that the outlet heaters were not raising the temperature of this 55 F air. The fact that this air coming in at 55 F raised the temperature over the playing floor to 59 F from 52 F in this one-half hour period from 8:00 to 8:30 p. m. seems due to its being heated by the audience. Some of the spectators were on wooden bleachers between the playing floor and balcony. The continual dropping of temperatures at higher levels during this period seems to substantiate the belief that low temperature air was being discharged into the room. The audience then, seems to account for this rise in temperature over the playing floor. It should be noted that the main body of air was dropping in temperature during this period.

This rise in temperature of air over the playing floor was seemingly checked by taking the entire air supply from outside from 8:30 to 8:50 p. m. The temperature dropped from 59 F to 55 F in 15 min. Since this outside air would be heated to 55 F in the tempering coil, just as was the partially recirculated air from 8:00 to 8:30 p. m., it should leave the outlet ducts at 55 F and cause a rise as before. The temperatures at the 25 ft or thermostat level were dropping and had reached 64 F at 8:30 when all outside air supply was started and they continued to drop. Probably then the air was leaving the outlets some of the time at considerably higher temperatures than 55 F. This would cause the air to rise and pass up through the balcony levels without influencing temperatures at the 5 and 15 ft levels over the playing floor.

The change to one-half recirculated air at 8:50 p. m. caused the air temperatures at the 5 and 15 ft levels to rise about 2 F in 15 min. All other temperatures increased about 1 F. The steam consumption increase from 9:00 to 9:15 p. m. is likely due to shower use.

The rise in average air temperature with 50 per cent recirculation from 9:00 to 9:15 p. m. and the drop with no recirculation from 8:30 to 8:45 p. m. seems to indicate a possible inadequacy of the heating capacity. The number of tempering coils turned on during this game is unfortunately not known. If only two were turned on, the capacity would be inadequate to heat all outside air to 55 F, but would still be capable of raising ½ or ¾ outside air to that temperature. This would aid the explanation of the results obtained.

The humidity curve rises and falls somewhat inversely as the temperature at the 15 ft level which indicates but little change in absolute humidity. The change seemed to be a decrease of 0.5 grain per cubic foot from 8:00 to 9:00 p. m.

The temperatures for the lower balcony and for the elevations the upper balcony will occupy when finished, averaged 66.4 F, 64.6 F, and 61.9 F at 8:00, 8:30, and 9:00 p. m. respectively. The corresponding variations from top to bottom of balconies was 4.5 F, 2.5 F, and 4.4 F.

Fig. 7 shows observations when taking in outside air in milder weather than that of Fig. 6. The attendance was 4000, about 800 more than that for Fig. 6.

The relative humidity was almost 50 per cent at 7:45 p. m. The taking of ¾ of the air supply for the fans from outside reduced the relative humidity to about 36 per cent by 8:30 p. m. The air change through open doors as the crowd came in from 7:45 to 8:00 p. m. aided in this moisture reduction. The reduction in absolute humidity was 0.9 grain per cubic foot in three-quarters of an hour. After 8:30 p. m. an equilibrim moisture content of 2.37 grains per cubic foot was maintained.

The temperatures remained practically uniform above the 15 ft level. The largest variation during the period of the game from 8:00 to 9:00 p. m. was 1 deg. The variation at the 15 ft level was only slightly greater than this. At the 5 ft level, a gradual increase in temperature took place in this period. Probably during this period the balcony thermostats kept the steam shut off from the outlet heaters most of the time. The air would then be discharged from the outlets at the fairly low temperature resulting from the mixing with recirculated air and from the heating in the tempering coils. From the fact that the 15 ft level temperatures dropped slightly while the 5 ft level temperatures were rising seems to indicate that the air reaching the playing floor was about 61 F. Some heat probably was picked up by this air from spectators as it passed through the wooden bleacher sections. When the damper positions were changed at 8:45 p. m. so as to take in only ¼ outside air and recirculate ¾ building air, the temperature did not rise at the 5 ft level but did rise about 1 deg in 15 min. at the 15 ft level. This appears due to the air being warmer at the points of discharge therefore rising above the 5 ft level over the playing floor area. The doors were opened at 9:00 p. m. which accounts for the checking of the rise in temperature.

General Discussion

In general, it may be said that many times the operation of the tempering units is not under thermostatic

control. Very often only one of the three coils in parallel is supplied with steam. The effect is then to make the area of the other two coils serve as a by-pass area not under thermostatic control. The air then must be quite warm as it reaches the coil location if it is to be heated sufficiently to cause the thermostat to open up the dampered by-pass. The effect is then to hinder thermostatic control under many conditions. The reason for leaving off one or two of the coils much of the time is to prevent overheating when recirculating. When recirculating and the by-pass damper is wide open, the temperatures would still be much above the desirable duct temperatures as established on the tempering unit themostats. An addition to this control, so that at the same time the by-pass damper is opened, a damper would close off the air passage through the coils, would obviate the necessity and undesirability of turning off the steam from one or two coils. Another variation for better control might be the placing of thermostatic valves on each of the three coil supply branches arranged so that they would open in succession.

Disturbance of Still Air Gradient by Fan Operation

Fig. 8 shows the results of fan operation on the still air temperature gradients on a cold day at a time when the building was not occupied. This was on the coldest day of the year. The mean temperature on this day was —15 F. On the previous day, it was —10 F and

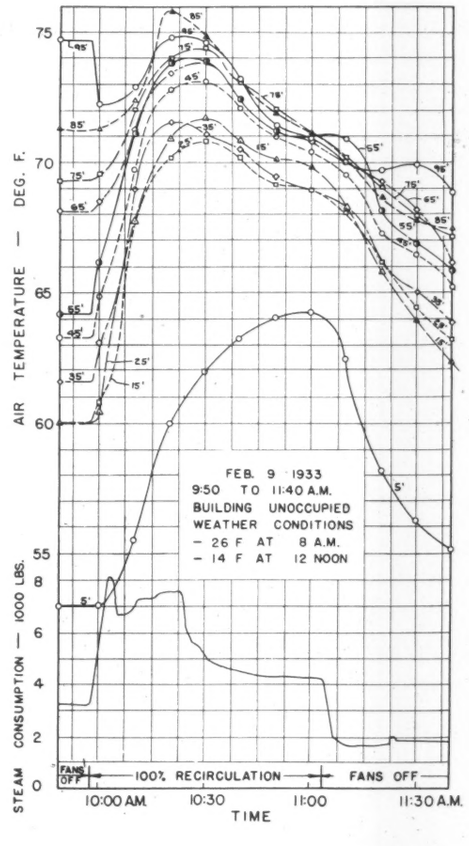


Fig. 8—Disturbance of still-air temperature gradient by fan operation in extreme cold weather

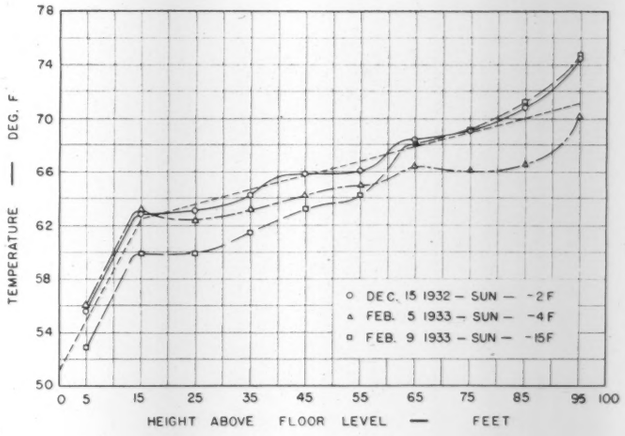


Fig. 9—Temperature gradients for gravity circulation on three cold days

on the day before this it was 0 F. The average wind velocities on the three days were 13.1, 11.6, and 18.5 miles per hour. At 8 a. m. the temperature was —26 F and at 12 noon, it was —14 F. The roof vents were closed during these days.

Readings of temperatures were taken at 9:50 a. m. from floor to roof under still air conditions. The temperature at the 5 ft level was 53 F; at the 15 and 25 ft levels, it was about 60 F. At the 85 ft level, it was slightly over 71 F and at the 95 ft level, it was almost 75 F. This was a spread of about 22 F. Of this amount, 7 F increase occurred in the first 10 ft and almost 4 F in the last 10 ft. This will be discussed further in connection with Fig. 9.

At 9:57 and 9:59 a. m., the fans were started with 100 per cent recirculation of building air. Readings taken at 10:00 a. m. show the influence of the fan circulation. The 95 ft temperature was lowered 2½ F. The temperature at the 55 ft level was raised 2 F. Above this level, the rise was about one-half degree. Below at the 35 and 45 ft levels, it was about 1½ F. The temperature at the 5 ft level showed no change. The readings were taken in quick succession from the bottom to the top, which influences the results somewhat since the time interval from the starting of the fans was not exactly the same for all levels. A quick response to changing temperatures on the part of the resistance thermometers is indicated and also a definite lowering of roof temperature and a raising of temperatures near the 55 ft level.

The steam demand reached a peak of 8000 lb upon starting the fans. The previous demand had been about 3200 lb per hour, which checks with the calculated heat loss of about 2,800,000 Btu for —15 F weather. The heavy demand for steam lasted for about 25 min. Since the recirculated air was well above 55 F, the by-pass around the tempering coils must have been open. The flow then would be divided between the tempering coils and the by-pass so that no great degree of heating would be done here. The air at the thermostat level of 25 ft was at a temperature of 60 F at the outset so that the outlet heaters were responsible for considerable heating. The temperatures at all levels continued to rise

until 10:30 although the 65 F temperature at the 25 ft level was reached at 10:10 a. m.

The temperature at the 5 ft level also increased, but more slowly than that at the higher levels. The temperature at this level continued to rise until the fans were shut off shortly after 11:00 a. m. The temperatures at all higher levels dropped during this half hour ending at 11:00 a. m. This was due to heat being supplied at a rate below that of the heat losses. The temperatures at the thermostat level were well above the 65 F that would permit steam to enter the outlet heaters. The main source of heat during this period would be the tempering coils. The steam consumption was about 1000 lb per hour higher in this period than it had been previous to fan operation. An inspection at 11:00 a. m. of the thermometers on the 24 balcony thermostats showed temperatures from 67 F to 69 F with most of them at 68 F.

The spread of temperatures at 11:00 a. m. was 2.2 F from the 15 ft to the 95 ft level. The difference between the 5 and 15 ft levels was 4.7 F. The total difference was about one-third that observed before the fans were started. Since the air at the 5 ft level is below the level of the air outlets under the balcony, a low discharge temperature should be more effective in heating this level than high temperature air. Perhaps an audience seated in the balconies would aid heating at this level by retarding the upward flow of warm air. Had the balconies offered the same resistance to air flow as would obtain when filled with spectators the difference in temperature of 4.7 F between the 5 and 15 ft levels would probably have been further decreased.

When the fans were shut off at 11:05, the difference between the highest and lowest temperatures increased. At 11:40 the total difference was almost 14 F, one-half of which was in the 10 ft between the 5 and the 15 ft levels. The curves if extended seem to indicate that still air conditions found previous to the start of fans would be obtained again shortly after 12:00 noon or about one hour after shutting down the fans. The steam consumption in this period was about 1600 lb per hour, which is about one-half of that previous to the fan operation. This likely is due in part to heat stored in the structure.

Temperature Gradients on Cold Days with Gravity Circulation

Temperature gradients observed under conditions of no occupancy and still air conditions on three cold days are plotted together on Fig. 9. The temperature data shown on Fig. 8 at 9:50 a. m. appears as the Feb. 9, 1933 points with —15 F outside temperature. The Feb. 5, 1933 temperatures were observed at a time when the outside temperature was —4 F. On Dec. 15, 1932, data were taken when the outside temperature was —2 F.

All three curves show an upward turn from 85 to 95 ft. Since there was bright sunshine on all three days, this was probably due to the sun effect. The rate of rise from 5 to 15 ft was considerably greater than that for the remaining distance in all three cases. The curve for —15 F weather has the steepest gradient. In each case the temperature at the 25 ft level where the thermostats are located was below that for which they were set. This shows that the steam would be on all coils and that under reverse gravity circulation the system lacked the capacity necessary to maintain conditions desired for occupancy periods. On the coldest day, this inadequacy resulted in about three degree lower temperatures at the lower levels than on the days with near zero temperatures.

The weather was not greatly different on Dec. 15 and Feb. 5. It seems impossible to account for the difference in gradient on these two days. The temperatures at lower levels were the same whereas at the higher levels this difference became about 4 F. The previous weather does not offer an explanation. On Dec. 13, 14 and 15 the mean temperatures were 6 F, 7 F and —5 F with minimum temperatures of —5 F, —1 F and —10 F respectively. On Feb. 3, 4 and 5, they were 16 F, 2 F, and 2 F, with minimum temperatures of 12 F, —8 F and —12 F. The mean of 2 F for Feb. 5 is not a true indication of the weather at the time of the readings for that day. The weather changed rapidly in the afternoon due to warm southwest winds. The temperature was —10 F at 7:00 a. m. and —2 F at 12 noon. The average wind velocities for the three December days were 11.0, 11.4 and 9.7 miles per hour and were 8.8, 11.4 and 11.4 miles per hour for the three February days involved. The roof vents were closed during all of these days. No explanation is known for the difference in these two gradient curves.

Since gradients under such extreme temperatures are useful for calculating the heat losses that must be met by the heating system under the severest conditions, the results for Dec. 15 are considered to show the design conditions for zero weather. A broken straight line has been plotted through the points, neglecting the sun effect. The straight line for the lower levels was intentionally lowered 1 deg that it might intersect the straight line for the higher levels at the 15 ft level.

The equation for determining the temperature at any elevation for gravity circulation from 0 to 15 ft would be

$$t = 51 + 0.76\,H$$

where
t is the air temperature at H elevation in feet
51 is the air temperature at the dirt floor.

The similar equation for temperatures at elevations from 15 ft to 100 ft would be

$$t = 51 + (0.11\,H + 9.7) \text{ or } t = 60.7 + 0.11\,H$$

where t, H and 51 are as above and 9.7 is the difference in intercepts on the temperature axis

The equations show that the rise in temperature is about ¾ of one degree per foot in the first 15 ft and about 1/10 of one degree per foot for the remainder of the height.

The difference in temperature from the lowest point of the lower balcony to the highest point of the upper balcony is about 5 F. It will be noted that the temperatures were practically the same at the 15 and 25 ft levels. Why there was not a gradual transition from the steep gradient of the lower levels to the slower gradient of the upper levels is not known. The thermometers were checked in order to account for this behavior but no error was found. Possibly the movement of air towards the outlets which act as inlets on reversed gravity flow causes this disturbance to a smooth

gradient transition. These outlets are located near the 15 ft level.

It would appear that the heating of air at the 15 ft level, the carrying of it through ducts to the 73 ft level, and there discharging it towards the peak of the building would be conducive to the maximum temperature gradient. Under these conditions of gravity flow, the fans not running, the system corresponds closely to a high outlet gravity convector heater system. The inlets are below the balconies at the 15 ft level and the main sources of heat are located here. Additional heat is added a short distance above this in the tempering coils. The discharge is at the 73 ft level through the 12 recirculating openings located at that elevation. Building air probably enters the lower recirculating inlets of the fan system located at the 58 ft level, flows upward and mixes with heated air which is discharged through the openings at the 73 ft level.

The location of the inlets of these gravity convectors 15 ft above the floor probably accounts for the abrupt change in gradient experienced at the 15 ft level. The location of the outlets at an elevation of 73 ft would appear to be conducive to high ceiling temperature.

Conclusions

The system as installed is considered to be well suited to the heating and ventilating requirements of the building. During inoccupancy periods, gravity circulation places a limit on the heating capacity and therefore the cost of heating. During occupancy, the balconies are heated by a high temperature discharge from outlets placed below them. When the desired balcony temperature is reached the steam is shut off from the outlet heaters and air is then discharged at the lower temperature desired at the playing floor and diffuses over the lower levels. When the air over the playing floor is warmer than desired, this entering air will lower the temperature. If it is cooler than desired, the entering air will raise its temperature.

Under gravity circulation and near zero outside temperature conditions, a temperature gradient of about ¾ of a degree per foot was found for the first 15 ft of elevation and about 1/10 of a degree per foot from 15 ft to 97 ft at the peak of the roof. While these results apply strictly only to a building and a heating system similar to that tested, it is considered that this gradient would be approached in any large single room building, exposed on all sides and on the ceiling and with heating units located 15 ft or less in height above the floor. Should radiant heaters be placed at higher elevations the gradient probably would be increased.

A fan system is considered desirable in this type of building in order to reduce humidity by the introduction of outside air and at times to remove the heat from the audience. The fan system when operated recirculating reduces the temperature gradient. On a —15 F day and no occupancy, the gradient of about 22 F with gravity circulation was reduced to approximately 7 F with fan recirculation. Just how much of a temperature gradient will exist under fan recirculation depends upon, among other factors, the size of the audience and the discharge temperature from the outlets.

It is considered safe and desirable in this type of a building that the designer use the temperature gradient observed under gravity circulation rather than any found with fan recirculation.

Society to Celebrate 40th Anniversary in New York

During the week of February 5, 1934, the Society will hold its 40th Annual Meeting at Hotel Biltmore, New York City, with members of the New York Chapter as hosts. Pres. H. L. Alt of the New York Chapter, has announced the appointment of Arthur Ritter as Chairman of the Committee on Arrangements and subcommittees to handle the reception, registration, entertainment, transportation and publicity are being organized.

During the same week, February 5 to 9 inclusive, the Third International Heating and Ventilating Exposition will be held at Grand Central Palace, New York.

Special sections have been reserved in the Exposition for gas heating equipment, oil burners, warm air heating, air conditioning, cooling and refrigeration. Reservations have been made by leading manufacturers of apparatus and all indications point to a comprehensive display of equipment required for any heating, ventilating or air conditioning service. The development of two years will be noticeable in the new types and improved models that will be demonstrated by the exhibitors.

An Exposition Advisory Committee headed by C. V. Haynes, First Vice President of the Society, is cooperating with the exposition management.

Construction Code Explained to New York Builders

Nearly 500 members and guests of the New York Building Congress attended a luncheon at Hotel Roosevelt, New York, August 16 to hear an address by Stephen F. Voorhees, Chairman of the Code Committee of the Construction League of the United States. Mr. Voorhees gave an analysis of the Code of Fair Competition for the Construction Industry and outlined the provisions of the blanket code which had been submitted under the provisions of the NRA.

In giving the history of this project Mr. Voorhees pointed out that the master code was sponsored by the Construction League and that other national associations were preparing supplementary codes to cover architecture, engineering, contracting and manufacturing of materials.

The speaker said that the plan of administration provides for the principle of self government in each group and urged everyone in the building field to support President Roosevelt's efforts.

Some of the New York members of the Society who attended were: Henry C. Meyer, Jr., John G. Eadie, R. Donnelly, W. W. Timmis, L. K. Berman, R. K. Raisler, W. L. Keplinger and A. V. Hutchinson, secretary of the Society.

Measurement of the Flow of Air Through Registers and Grilles*

By **L. E. Davies†, Chicago, Ill.**

NON-MEMBER

This paper is the result of research conducted at Armour Institute of Technology of Chicago, in cooperation with the A. S. H. V. E. Research Laboratory and the Ventilating Contractors Employers Association of Chicago

THE results obtained in the laboratories of Armour Institute of Technology from tests conducted to determine the most accurate and simple method of measuring the air flow through registers and grilles have been presented to the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS at two previous meetings. As it appeared that the only instrument that was practical for use in this manner was the anemometer, the work resolved itself into a study of that instrument and its characteristics. It must be clearly understood that there was no expectation of developing a precise laboratory method, but rather a method applicable to field tests where errors not to exceed 5 per cent may be considered reasonable.

The investigations were conducted with both supply and exhaust grilles. The setup consisted, briefly, of a well constructed system of ducts so arranged that the same air that passed through the grilles was also drawn or forced through a straight round section of duct of relatively small cross-section to insure rather high velocities. Pitot tube traverses, taken in two directions across this pipe, were used as the basis for comparison with anemometer traverses taken by various means at the face of the grille. This test setup, as arranged for supply grilles, is shown in Fig. 1.

To prevent confusion the discussion will be confined to supply grilles.

In the first paper[1] a new formula was developed, namely:

$$\text{cfm} = \frac{C_v V (A + a)}{2}$$

in which

cfm = volume of air in cubic feet per minute
V = average indicated velocity obtained by an anemometer traverse taken in contact with grille face (after applying the usual instrument correction)
A = gross area of grille, square feet
a = net free area, square feet
C_v = coefficient obtained by experiment

The virtue of this formula was that this coefficient was very close to unity, so for many practical purposes, it could be omitted entirely.

The observations upon which this formula was based were of a rather limited character, being confined to thin metal grilles of a simple lattice work design, except for one observation with a piece of expanded metal lath. One type of anemometer was used throughout the early experiments, and most of the observations were taken with fairly satisfactory types of approach ducts. Since that time an attempt has been made to determine how to apply this formula to the many variable conditions encountered in practice. These conditions include:

1. Different types of anemometers
2. Grilles of ornamental design
3. Registers of various special and unique design
4. Unit heater cores and unit ventilators
5. Various types of approach ducts
6. Grilles of various dimensions
7. Lattice work grilles in which the frets are relatively large
8. Grilles backed by screens, heater cores, adjustable dampers, etc.

Some of these cases were discussed in a previous paper,[2] particularly the use of the different types of anemometers. The most important disclosure brought out in this second report, however, was the fact that when the formula was applied to two special grilles of an ornamental plaster design, the results were very unsatisfactory, the error in one case amounting to 20 per cent while in the other it reached a peak value of 45 per cent. Recently considerable time has been devoted to an attempt to explain this phenomena with results that will be discussed in detail elsewhere in this report.

In order to have a basis for comparison a certain set

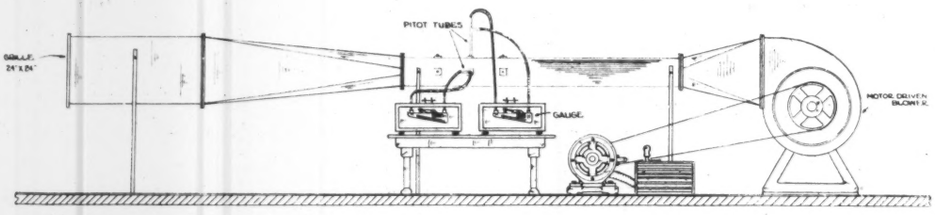


Fig. 1—Test set-up for supply grilles

*Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Hotel Statler, Detroit, Mich., June, 1933.

†Associate Professor of Experimental Engineering, Armour Institute of Technology.

[1]The Measurement of the Flow of Air Through Registers and Grilles, by L. E. Davies, A. S. H. V. E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, January, 1930.

[2]The Measurement of the Flow of Air Through Registers and Grilles, by L. E. Davies, A. S. H. V. E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, April, 1931.

of conditions will be assumed, and on the basis of an earlier report these conditions will be designated as normal. The specifications for these normal conditions are given in Table 1.

Table 1—Specifications for "Normal Conditions" in using anemometer to measure air discharged from a grille

Type of grille—Plain lattice work of thin metal.
Free area—30% to 90%
Overall dimension, ft—2x2
Width of fret—¼ in.
Thickness—$\frac{1}{16}$ in.

Type of approach—Straight duct full size of grille, free from elbows, dampers, splits or other forms of obstruction.

Character of discharge—Straight forward—uniform distribution.

Character of anemometer traverse—16 spot readings taken at centers of 6 in. squares, anemometer in contact with grille face.

Type of anemometer—4 in.—200 ft to 3000 fpm capacity.

Formula to be used—cfm $= \dfrac{C_v V(A+a)}{2}$

Values of C recommended.

Air Velocity, fpm	C_v
150	0.952
200	0.957
300	0.967
400	0.977
500	0.985
600	0.992
700	0.998
800	1.000

Now it is apparent that if any one of these conditions is changed the results obtained may be in error. For example, if a condition existed in the approach duct which caused the air to be discharged at a pronounced angle it might conceivably cause the anemometer to read low. If this were a fact, and the normal procedure and normal formula were used, the calculated volume of air would be lower than the true volume by an amount proportional to the error in the anemometer reading. This error might be corrected for either by some modification in the standard procedure, or by the application of special coefficients or correction factors.

Considering the number of variable factors it might appear then that it would be necessary to use a veritable book of coefficients and special instructions to deal effectively with the various cases. Fortunately, this is not true. Several of the factors are of little consequence and others can be handled by the application of a few simple rules of procedure.

Let us now consider some of these factors and their effect on the accuracy of the results obtained when the normal procedure and normal formula and coefficients are used.

Different Types of Grilles

Considering, first, the effect of changes in the design of the grille itself, Fig. 2 shows some of the grilles that have been tested and found to give results identical with those obtained with the plain lattice work type. It will be noted that the frets in the different types differ considerably. For example, *f* has the small round wire, *c* and *d* thin flat strips half an inch deep, *e* diamond shaped, *a* rather deep tapered frets, etc. In addition to those illustrated several others of similar character have been used, including the lattice work designs with various widths of frets under ⅜ in., and various percentages of free area. Tests were made using ornamental iron and plaster designs where the figures in the

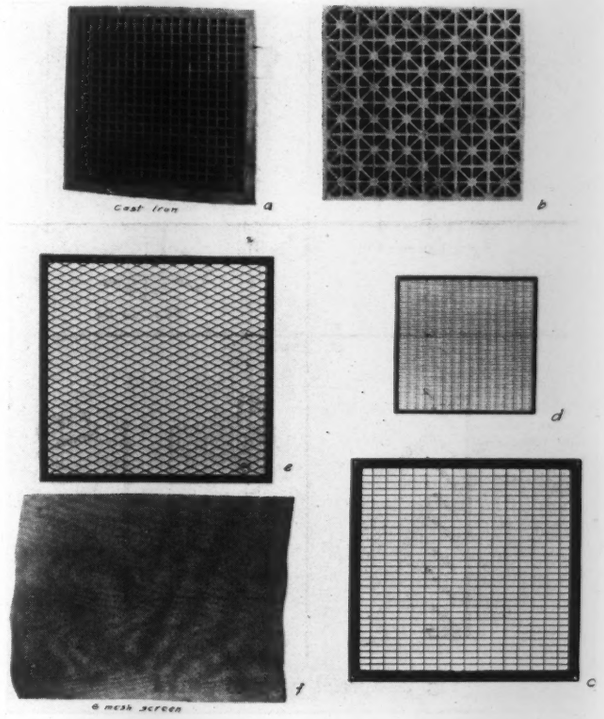


Fig. 2—Examples of grilles which do not require special treatment

design were all of relatively small size. Tests also were made using grilles of various overall dimensions and proportions, from approximately 9x12 in. and 5x40 in. to the 24 in. square that was used in most of the early work. So far as the design and size of grilles is concerned, the conclusions are that the normal procedure can be followed within the following limitations:

1. Size—anything except possibly those of extremely small dimensions.
2. Percentage of free area—no limitation
3. Design—either plain or ornamental
4. Depth of grille—any depth not to exceed approximately 10 times the minimum distance between frets.
5. Width of frets—not to exceed ⅜ in.

It should be noted that throughout this paper the term *grille* will be used in a rather broad sense so that it may refer to anything having the general character of a grille, such for example, as radiator cores. Thus, in Item 4, are called to mind certain heater cores of the tube and fin type, where the depth is several inches, and the distance between fins only a fraction of an inch. In such cases the resistance in the long narrow passages has the effect of a much smaller free area than the actual measured value. Hence, if the standard procedure is followed, the calculated air flow may be some 10 to 15 per cent too high, even when the velocity distribution is uniform. With an irregular velocity distribution such as is obtained when the propeller or disc fan is used, the error may be much greater.

In some instances grilles of the usual types are backed by ordinary wire screen. Tests were made to determine how to handle these situations. The standard procedure can be followed exactly except that in determining the free area both the grille and the screen must be con-

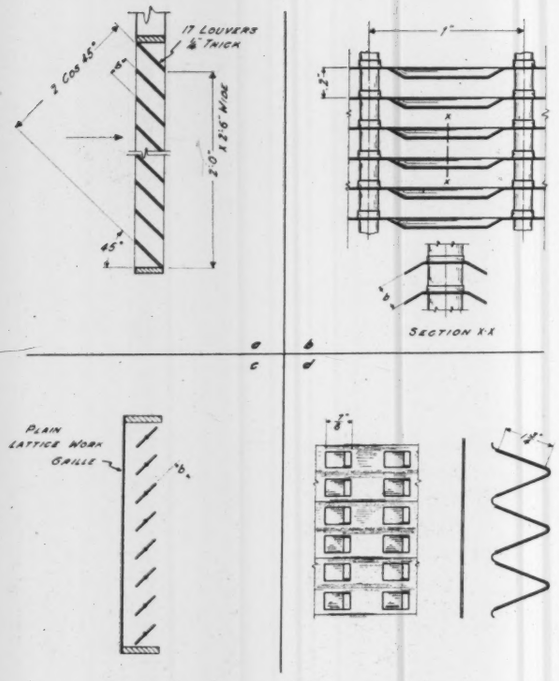


Fig. 3—Special types of grilles

sidered. The percentage of free area of the combination, where they are in reasonably close contact, can be taken as the product of the two individual values. Thus, for example, imagine a plain lattice work grille, 4 square feet in area and with 60 per cent free area backed by an 8 mesh screen having 70 per cent free area. The free area a to be used in the formula would be:

$$a = 4 \times 0.6 \times 0.7 = 4 \times 0.42 = 1.68 \text{ sq ft}$$

If the screen is several inches removed from the grille proper, it will fall into a class to be discussed later.

Grilles Requiring Special Consideration

Certain types of grilles (using the word again in its broad sense) are of such unique design that there naturally exists in the mind of the operator considerable doubt as to what should be taken as the free area. Examples of some of these are shown in the sketches *a, b, c, d,* of Fig. 3, and the photograph, Fig. 4.

Sketch *a* in Fig. 3 illustrates a door louver similar to the one used by Professors Larson and Nelson in their Investigation of Air Outlets,[3] and Sketch *b* is a grille of the same general character as the one illustrated in *d* of Fig. 2, except that the outer edges are turned down about 30 deg.

The general rule in all these cases is to take for free area the *minimum* cross sectional area at any point in the air stream. Where the frets are at an angle this area will be the area between them measured normal to their surface (distance *b* on sketches). As an example of how this rule is applied, consider the door louver illustrated. The anemometer traverse should be made across the face of the louvers in the usual manner, the upper strip through which no air can issue being, of course, not considered as part of the face. The gross area, A, in the formula should always be taken as that area over which the traverse is taken, in this case 5 sq ft. The free area would then be taken as

$$a = 2\tfrac{1}{2} \times (2 \cos 45° - 16 \times \tfrac{1}{4} \times 1/12) = 2\tfrac{1}{2} \times 1.081 = 2.702 \text{ sq ft}$$

The same result would be obtained if the free area at the face were simply calculated and multiplied by the cosine of the angle of the louvers. Or many will prefer to caliper directly the minimum distance between the louvers. The actual method of calculation for free area is optional but in cases of this kind the problem must be studied carefully for it is easy to make an error.

When Professors Larson and Nelson ran their anemometer calibration tests on the door, they found it necessary to use a coefficient of from 0.78 to 0.85 in the $\frac{C_v V(A + a)}{2}$ formula. They used the free area measured at the face. They furnished complete copies of the original data, and when recalculated by the method here described the average coefficient obtained from thirteen runs differed from those which have been recommended by only slightly over 1 per cent, the maximum discrepancy being 5 per cent, obtained at a very low velocity. These results are regarded as of particular

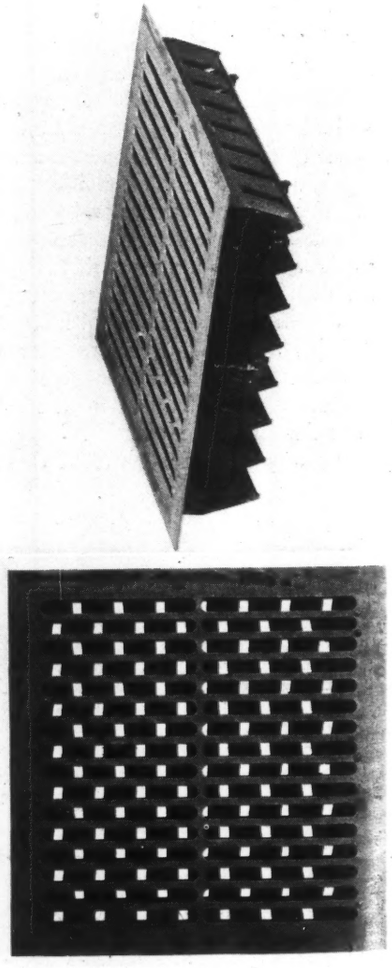


Fig. 4 — Grille of unusual design

[3]See Investigation of Air Outlets in Class Room Ventilation—A. S. H. V. E. Journal Section, *Heating, Piping and Air Conditioning,* June, 1932.

value because of the fact that they were obtained by a different experimenter and with different apparatus.

Another application of the same principle is with the grille shown in Sketch *b*, Fig. 3. As already stated, this is identically the same as the one shown in photograph *d*, Fig. 2, except for the turned down edges. These two types are classed by their maker as straight fin, and angle type fin respectively. The manufacturer lists these grilles as having 84.3 per cent free area, and careful measurement on the straight fin type was made to verify this figure. When traverses were made on the straight fin type by the normal method the results obtained were very satisfactory. With the angle type fins, however, the calculated quantities were some 6 per cent to 7 per cent high. A careful check on this type showed that when the free area was calculated by taking the minimum distance betwen the fins (at the point where they slope downward, *b* on sketch) it was found to amount to only 76 per cent of the gross area. Using this smaller figure in the formula the error in the anemometer traverse was reduced to less than 2 per cent.

The sketches, *c* and *d*, of Fig. 3, illustrate two registers with which interesting results were obtained. The first of these was a plain lattice work grille, behind which was a set of dampers. Runs were made in the usual way with the dampers set for various angles. The values of A and a in the formula were first taken directly from the grille face, the presence of the dampers being ignored. It was found that accurate results were obtained up to the point where the minimum area through the dampers (calculated by the same means used with the louvers already discussed) became less than the free area of the grille itself. Beyond that point the calculated results were too high unless this minimum free area through the dampers was used for free area in the formula.

The second of these registers is of special interest because of its unusual design. Its face may be either the usual lattice work or the bar type shown in Fig. 4. In back of this face grille is a zig-zag series of shutters consisting of two thicknesses of metal lying flat against each other. Each of these metal strips contains rectangular openings measuring ⅞ in. by 1¾ in. so placed with respect to each other that as the shutter operates the ⅞ in. dimension can be reduced by any desired amount. Traverses were made in the usual manner with the shutters in various positions, and in each case calculations were made in two ways, first, by using the free area of the grille and secondly, by using the area through the shutters. The results are shown in Table 2.

Table 2—Results obtained from traverses made on register shown in Figs. 4 and 3*d*

Position of Shutters	Gross Area Sq Ft	Net Free Area Through Grille	Net Free Area Through Shutters	Per Cent Error Using Grille Area	Per Cent Error Using Shutter Area
Full open	1.614	0.977	1.19	+ 7.0	+17.5
¼ closed	1.614	0.977	0.935	+ 9.0	+ 7.0
½ closed	1.614	0.977	0.595	+12.0	− 4.7
¾ closed	1.614	0.977	0.2975	+32.0	− 2.5

Here again it will be seen that when the shutters were wide open the greatest accuracy was obtained by using the grille area, which is smaller than the shutter area, whereas when the shutters were partly closed so as to reduce their area to a figure smaller than the free area of the grille, the greatest accuracy was obtained by using the shutter area. The results were somewhat less consistent than those obtained with plain grilles but this was to be expected under the circumstances.

Other observations taken with one grille mounted two or three inches in front of another, or in front of a heater core, indicate that the rule discussed here may be taken as a general one. That is, when a grille has behind it, within a few inches distance, any obstruction of the general character of those described, the gross areas should be measured at the outer face where the anemometer traverse is made, but the free area should be taken at that point in the air stream at which it is a minimum.

The fact has been mentioned that when runs were made on two ornamental plaster grilles by the usual method some very large errors were found. No explanation for this phenomena was offered in the earlier paper but further thought and experiment since that time has produced gratifying results.

In order to find the explanation for the inconsistent results obtained with these grilles it was necessary to analyze them carefully in order to determine in what manner they differed from the other grilles. This analysis shows the following characteristics:

1. The grilles were quite thick.
2. The designs were of an ornate character.
3. The surfaces were rough and not all in one plane.
4. The outer edges were rounded off.
5. The details of the design were large, that is, large irregular frets and large interstices.

The first four of these factors were quickly eliminated by the fact that satisfactory results were obtained with other grilles possessing one or more of these characteristics. This directed particular attention to the last factor, because the one having the largest details and widest frets, gave the poorest results. Furthermore, when traverses were made with the 6 in. anemometer the results were considerably improved, there being a much greater difference in the readings of the two instruments than is normally the case.

In order to study the subject in a systematic manner a large number of lattice work grilles were made up out of strips of wood, approximately ⅛ in. in thickness and of various widths up to 1¾ in. and with various spacings so as to give, in each case, several amounts of free area. Traverses were made in various ways, to be discussed later, and with various types of anemometers. It was disclosed that as the size of the fret increased the error obtained by the standard formula also increased, reaching a value of over 40 per cent with a 1¾ in. fret, 30 per cent free area and the 4 in. anemometer. This means that in the case cited, it would be necessary to introduce into the formula a new coefficient having a value of approximately 0.7. It appears then that when the frets of a grille are large, it will be necessary to add a second coefficient to the normal formula, making it read

$$\text{cfm} = \frac{C_f C_v V(A+a)}{2}$$

in which C_v is the normal coefficient, recommended for grilles having small frets, and C_f a special coefficient for

use only when the frets are large. Under the various circumstances this value of C_f will range from 0.65 to 1.00, being a function of three factors, namely, percentage of free area, size of fret, and size of anemometer used. These last two factors may be considered as one as it is really the ratio between the size of the fret and the diameter of the anemometer that determines the value of the coefficient, other things being equal.

The curve, Fig. 8, shows the values of C_f to be used for various widths of fret, and percentages of free area. Greater difficulty has been experienced in securing entirely consistent results in these cases than in any others dealt with. For this reason the curves are being submitted as tentative, subject to possible revision as additional data are secured. However, they may be used with the assurance that under the conditions to which they apply, the results obtained will be more accurate than would be secured by any previously recommended method. It is planned to continue the research on this point, and a set of curves in final form will be submitted as soon as they can be secured.

As an example of the application of this method, reference to the results secured with the tests on the large plaster grille indicates that the frets of this grille varied from ¾ in. to 3 in., but a careful estimate of the average would place it at about 1½ in. The free area was 34.2 per cent. Consulting the curves it is found that the proper coefficient is 0.72. The results calculated by the normal formula were, at the higher velocities, 34 per cent too high. In other words, the calculated flow was 1.34 times as great as the actual flow. If this figure is multiplied by the new coefficient the result is:

$$\text{cfm} = 1.34 \times 0.72 = 0.965$$

or an error of 3½ per cent.

It should be emphasized again that this new coefficient need not be considered when the frets are less than ⅜ in. with the 4 in. anemometer, and ⅝ in. with the 6 in. instrument. The greater the size of the frets the more important it becomes to use great care in making the traverse. In all cases where the frets average over ¾ in. it is recommended that some one of the forms of moving traverse be used.

Methods of Making Traverse

Several methods may be followed in making the traverse to obtain the average velocity. These methods may be classified as follows:

1. Spot traverse—stationary readings.
2. Overall moving traverse.
3. Spot traverse—moving readings.
4. Strip traverse.

The first of these methods was used throughout the early part of the research, although enough readings were taken by the other methods to show that with proper care they all gave substantially the same results for the cases then under investigation. With the spot traverse method the grille face is divided into a number of equal rectangles having their least dimension approximately equal to, or greater than, the diameter of the instrument to be used. A reading of not less than 30 seconds, and preferably of one minute duration, is then taken with the anemometer held at the center of each rectangle in succession. Where the grille is very large the size of the individual rectangles may be increased so that not to exceed twenty such readings are taken. If the velocity distribution is reasonably good, it is permissible to use a smaller number of readings. In laying out these rectangles it is essential that they should not coincide with any repeating figure in the grille design except of course, very small figures such as the squares of a small lattice work grille. For example, suppose a grille is 24 in. square and the design is made up of a six inch square pattern repeated four times in each direction. If either of the spot traverse methods are to be used, the face should be divided either into 9 squares, 25 squares or 15 rectangles each 8x4.8 in. This principle should be followed even though it necessitates using dimensions so small that the successive anemometer positions overlap each other slightly. The larger the detail of the pattern, that is, the wider the frets and interstices, the more important this rule becomes.

The spot traverse is disliked by many workers because of the time involved. With these the overall moving traverse is preferred. Used with proper care this method may be very nearly as accurate as the spot method and under certain special circumstances may even become the most accurate. The usual procedure in using this method is to move the instrument along the upper edge of the grille, from one side to the other, then drop it down a short distance and return along a parallel path. This back and forth movement is continued until the entire face has been covered.

To secure satisfactory results two precautions must be observed. In the first place, the motion must be at a very steady rate, especially in those cases where the velocity changes a great deal from point to point. To maintain a uniform rate of movement it is advisable to figure out in advance exactly where the instrument should be at the end of certain time increments. For example, if it is desired to make a four minute traverse on a 2 ft square grille by making four horizontal strips across the face, the operator should keep one eye on the watch and constantly adjust his speed so that each successive 6 in. distance will be covered in 15 seconds. The second precaution is to allow for sufficient time. Here again, caution is essential in those cases involving rapid and large changes in the air velocity, whether they are due to the character of the approach or to peculiarities of the grille design. It is recommended that the instrument should be moved at a speed not greater than 2 ft per minute, and the duration of the run should be not less than *two* minutes. However, the personal element will always be a more important factor with this type of traverse than with any other, hence in general, it is not recommended where the greatest accuracy is desired.

The third method is similar to the first and differs from it only in that the anemometer is moved around slowly within the boundaries of the individual rectangles or squares. It is unnecessary to say, of course, that it is only applicable to those cases where the small rectangles are larger than the anemometer being used. With this method it is permissible to use fewer readings taken over larger rectangles. In effect this third plan is

compromise between the first two methods, and a very satisfactory one at that.

The fourth and final traverse method is really only a modification of the third. The grille is divided into parallel strips having widths approximately equal to the anemometer diameter. These must be such width, however, that they do not coincide with any repeating pattern in the grille design. The instrument should then traverse each of these strips from end to end, a separate observation being taken for each strip. If the strip is wider than the instrument it is advisable to zig-zag it back and forth as it moves along so that it follows roughly a repeating sine curve as the traverse is made.

When the grille design is made up entirely of small details and the air flow is distributed well out to the edge of the grille on all sides it makes little difference which of these methods is used. When the grille has very large details, however, carelessness in making the traverse may introduce very large errors.

As an illustration four grilles, 24 in. square were made up of wood strips 1¾ in. wide. These four grilles consisted of 3, 4, 5 and 6 strips respectively, in each direction. Traverses were made by the first method using 16 squares. The results obtained with the 3, 5 and 6 slat grilles were consistent both with respect to each other and with respect to other similar grilles having frets of smaller size. But the four strip grille was out of line by approximately 30 per cent, due to the fact that for each reading taken, the anemometer was in exactly the same position with respect to the strips; that is, in each case the center of the instrument was directly opposite the intersection of two frets. When the 4 in. instrument was moved about within the 6 in. square but slight change was noted. However, when nine observations were taken on 8 in. squares, and when the strip traverse was used, using 5 strips, the results were quite satisfactory. For this type of grille, and this includes the ornamental iron and plaster with large details, it is recommended that the strip traverse, or the spot traverse with moving readings, be used. Above all, be certain that the unit areas, whether they be long strips or squares, do not coincide with repeating patterns in the design.

Effect of Approach

The nature of the duct as it approaches the grille will naturally effect the character of the air stream.

The approach may cause the air to be discharged at a pronounced angle to the axis of the grille, or it may cause a very bad velocity distribution. These two effects may be present either independently or simultaneously. In general, a grille possessing considerable depth in proportion to the size of the openings, such as those shown in Fig. 2, *a* and *d*, will tend to straighten the air stream so that it will leave the grille at a 90 deg angle. They will not be particularly effective however, in causing a uniform velocity distribution. Other things being equal the most uniform velocity distribution will be attained with a grille having a relatively small percentage of free area.

The fact that the airstream strikes the anemometer at an appreciable angle to the normal does not in itself cause any serious error until the angularity becomes extreme. E. Ower, in his book *Measurement of Air Flow* states that an angle of 20 deg only causes an error of 1 per cent. He also reports an error of 19 per cent for 40 deg inclination. Recent experience with angular discharge from plain lattice work grilles on vertical risers showed only about 5 per cent error for a 40 deg angle of discharge. However, some portions of the air stream were at a smaller angle, which, no doubt, partly explains the relatively small error. Furthermore, the angular discharge is usually accompanied by a velocity change from a low value at the bottom to a peak value at the top. These two effects are, in part, compensating influences which also tend to reduce the error.

Considerable time was spent investigating the effect of poor velocity distribution on the accuracy of the traverse.

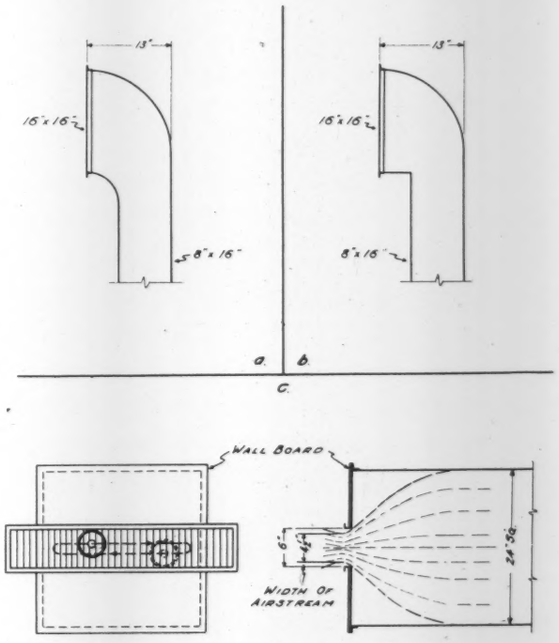


Fig. 5—Effect of approach on air stream

The vertical riser shown in section *a* Fig. 5, was one of those used to create this condition. It can be seen that it enlarges rapidly from an 8x16 in. stack to a 16 in. square grille. Several types of grilles were used. The discharge from the plain lattice and other thin grilles was upward at angles up to approximately 40 deg. The discharge from the grille shown in section *d* Fig. 2, was straight forward, while from the one shown in section *b* Fig. 3, it was downward at approximately 30 deg. These latter grilles, however, had a very large percentage of free area, and as a result the air was discharged at a high velocity at the top, while no air at all was flowing from the bottom portion. In no case was there any backflow.

The second riser, *b* in Fig. 5, had a sharp corner for the purpose of creating a reversed flow along the bottom, as this condition is encountered so frequently in practice. Although this did cause the backflow through a strip 1¾ in. wide with some of the grilles and a turbulent strip for another 2¾ in. the results, when taken by the normal method of 16 spot readings, four of which were negative, were surprisingly accurate considering the circum-

stances. The greatest error in any case with these two risers was 10 per cent, while the majority of the readings were accurate to within 5 per cent. However, from time to time special cases have been encountered where the error was excessive, so it is recommended that in all cases where a quiescent region or reentering air is encountered special precautions should be taken. Where there is simply a dead region this portion of the grille should be marked off and the traverse taken only over the remaining portion through which air is actually being discharged. Needless to say, only this latter part of the grille should be used in calculating the gross and net areas.

Where there is reentering air the problem is somewhat more confusing. If it is only at a very low velocity and over a small area, it may be neglected entirely, and the area treated as though it were in a quiescent state. If the amount of reentering air is large, however, it must be taken into account. To do this it is recommended that the grille face be divided into two parts, one carrying outcoming air, the other reentering air. The outcoming air should be measured by the usual method using whatever type of traverse is desired. The other portion should then be measured in the usual manner recommended for exhaust grilles. The volume of flow obtained by this second calculation should then be subtracted from the first to obtain the net volume. In many cases this procedure is not practical because the width of the strip through which the reversed flow takes place is too narrow to permit of an anemometer traverse. In such cases the best procedure will be to cover that

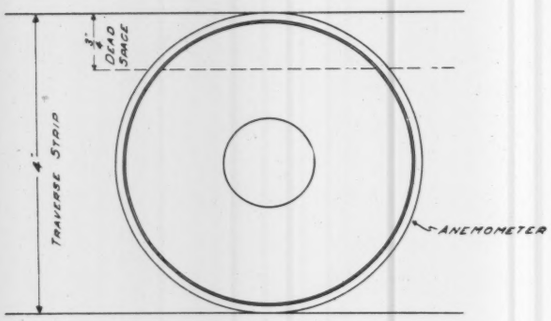


Fig. 6—Effect of dead strip

portion of the grille with card-board or adhesive paper, so as to prevent the inflow during the time that the balance of the grille is being traversed. This method has been tried with very satisfactory results.

As has been suggested before, even if these special methods are ignored and the traverse taken by the standard procedure over the entire face the results will not usually be as inaccurate as might commonly be supposed. This is only true, however, when the grille is approximately square and not too small. If one of the dimensions is quite small, say 4 to 6 in. the possibility of serious error is great because of the fact that a narrow strip along one edge may represent a very appreciable percentage of the total area. To illustrate this point a run was made with a long slender grille taken from the top of a unit ventilator. Not having an approach made to fit it exactly, it was placed across the end of a 24 in. duct and the remaining portion was blocked off with wallboard, as illustrated by *c* in Fig. 5.

Traverses were made by moving the anemometer very slowly from one end to the other while zig-zagging it up and down, and also by moving it first along the upper edge and then returning along the lower edge as indicated in the sketch. In both cases the calculated air volumes were 30 per cent too large. A careful study of the air stream was made with pitot tubes, small streamers and by holding the anemometer to one side of the grille and slowly moving it into the air stream until motion of the vanes was first observed. These observations disclosed the fact that due to the character of the approach and method of mounting, the grille was acting like any normal orifice and causing a contraction in the air stream so that the actual width of it in the plane of the anemometer vanes, was only 4½ in. When a traverse was taken using only the center strip the results were entirely satisfactory. Later another approach was obtained to fit this same grille. The contraction was then eliminated and accurate results were obtained with traverses taken over the full 6 in. width.

If the dead region along one edge is of considerable width it will not usually cause serious error even though it is entirely ignored and the traverse taken by the usual method. The anemometer readings along that edge will be very low, or even negative in value, and will thus reduce the calculated average velocity by the proper amount to compensate for the dead area. When, however, the dead strip is very narrow, say 1 in. or less, it will have practically no effect in reducing the speed of the anemometer. One reason for this can be seen by studying the sketch, Fig. 6. This shows a 4 in. anemometer traversing a 4 in. strip, in which the outer ¾ in. is dead. This dead strip represents almost 20 per cent of the traverse area but only 13 per cent of the anemometer area is exposed to it. Added to this fact is the inherent characteristic of anemometers, that, if there is a variation in velocity across their face, they tend to give a reading somewhat higher than the true average velocity.

Still another factor enters into the case of these slender grilles. Considering again the case of the 6 in. grille already discussed, Fig. 5, *c* shows two positions of the instrument and the course followed in making the traverse. It should be noted that the center strip, where the highest velocity is liable to exist is really

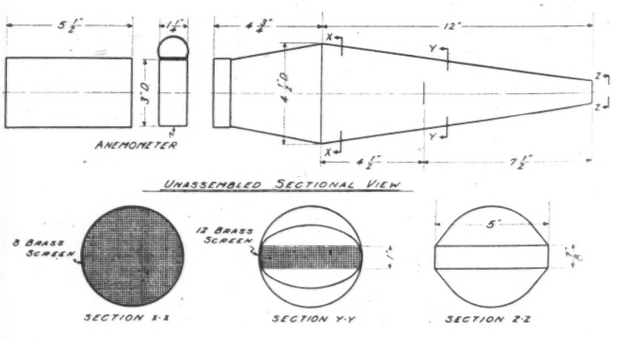


Fig. 7—New instrument for measuring air flow

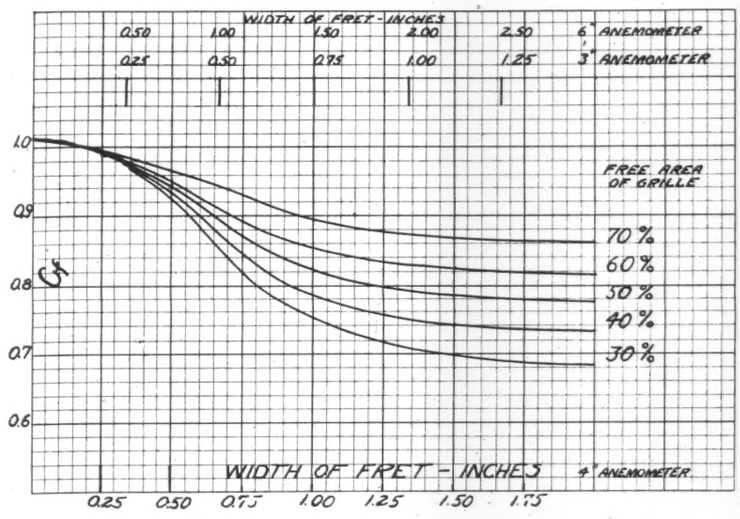
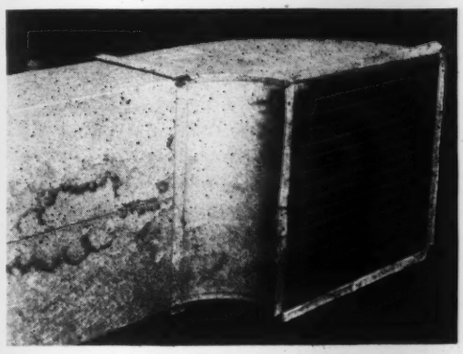


Fig. 8 (left)—Curves showing values of C_f for various widths of fret and percentages of free area

Fig. 9 (below)—Showing one type of approach tested



counted twice and thus has a disproportionate effect upon the final indicated average velocity. In these cases, that is, narrow grilles with rapid variations in velocity from side to side, it is recommended that one of the smaller types of anemometers be used.

A narrow strip of dead space will cause more error than a wider one. Probably the worst case of all is where there is air issuing from such a narrow strip at a very low velocity relative to the remainder of the stream. A dead strip or reentering air is readily detected and can be corrected for, but a narrow low velocity strip may very easily be overlooked. It is recommended that in all cases, the edges of a grille should be carefully studied before proceeding with a traverse in order that any unusual condition may be observed and a correction made.

As strange as it may appear one of the conditions under which it is easiest to go wrong is in making a traverse across the end of an open duct. When such a traverse was made across a 24 in. duct the result was some 20 per cent too large. This was due to the fact that with the grille resistance removed the jet coming from the 10 in. duct farther back, did not diffuse enough to completely fill the 24 in. outlet. A small dead strip, about 1 in. wide, existed all around the opening, representing 20 per cent of the outlet area and yet not wide enough to effect the anemometer readings. Placing a screen diffuser in the duct about 4 ft back of the grille corrected this condition completely and permitted a check between the anemometer and pitot traverses.

Table 3

Ventilator	Air Delivered Through Heater Core		Air Bypassed Around Heater	
	Per Cent Error Using $\frac{V(A+a)}{2}$	Coefficient Needed to Give Accurate Results	Per Cent Error Using $\frac{V(A+a)}{2}$	Coefficient Needed to Give Accurate Results
No. 1	+31.	0.764	+58.	0.634
No. 2	+36.	0.736	+73.	0.578
No. 3	+42.	0.705	+73.	0.578
No. 4	+46.	0.688	+42.	0.703
No. 5	+29.	0.774	+43.	0.700

Notes: Results obtained with 3 in. anemometer and unheated air. Readings taken with 4 in. anemometer averaged 2 per cent more accurate.

Unit Ventilators

Considerable time was spent in attempting to find a satisfactory means of measuring the discharge from unit ventilators, that could be used in testing these devices in the field. For calibration purposes a long straight circular duct was connected to the suction side of each unit, as shown in Fig. 10. Pitot tube traverses were made in this entrance duct, and compared with anemometer traverses taken by various means at or near the face of the grille. Observations were made on five different types of units submitted by the various leading manufacturers.

Runs were first taken by the usual methods, both spot traverses and the various types of moving traverses. Very little difference was apparent with the different methods. When the volumes were calculated by the $\frac{V(A+a)}{2}$ formula the average results were as shown in Table 3. It can be seen that the errors were very large, although quite consistent for those runs where the air was being passed through the heater cores. It would be possible, in these cases, to introduce a coefficient of 0.73 for the 3 in. instrument in which case the maximum error would be about 5 per cent. This is not to be recommended however, as it would be but a makeshift solution at best and with no assurance that it would continue to give even a moderate degree of accuracy with new designs that may be brought out in the future. In fact, Professors Larson and Nelson in their work, checked the figures very closely for one unit, but differed considerably with another. The similarity in the values of the coefficient were merely due to the fact that there was a general similarity in the types of construction in the different units.

In explaining the reason for the nature of the results obtained it must be pointed out that the grilles were all of a very open character having from 85 to 91 per cent free area while only a short distance below them were heater cores having much smaller areas. Thus, these heaters had far more influence in determining the character of the air stream than did the grilles. Applying the principle, which has already been discussed, that the

minimum area in the air stream should be used for net free area, *a* in the formula, the areas through the heaters should in this case be used. This method, has been used with quite satisfactory results. However, the fact is recognized that it is difficult and sometimes impossible to measure or estimate accurately the free area through most heater cores. Furthermore, the flow from most of these units was quite badly distributed, especially when the air was bypassed around the heater. In fact, in the latter case, the extremely poor distribution offers the only explanation possible for the very poor results. Hence, it must be concluded that it is impractical to make direct traverses at the outlet grille of a unit ventilator with the ordinary anemometer and be able to view the results with any degree of confidence.

There has always been a tendency on the part of some experimenters to handle situations like this by placing a duct over the outlet and then traversing the open end with an anemometer. This method should always be viewed with suspicion. If the duct is straight it must be very long in order to be certain that the air stream will completely fill it. Tests were made with straight rectangular ducts, equal in dimensions to those of the outlet grille. Results obtained were 20 to 25 per cent too high although the lengths of the ducts were some eight times the least dimension of the grille. It was necessary to taper the duct until the outlet was only about 70 per cent as great as the grille before anemometer traverses agreed with the true flow as shown by the pitot tubes. As it would be difficult to establish any fixed rule governing the amount of taper to be used, under different circumstances, and as there is always the probability that the presence of the duct will itself alter the flow, this method has never seemed to be a reasonable one.

The only solution to the problem appears to be in the development of some new device that will be relatively unaffected by non-uniform velocity distribution and which will not call for a knowledge of the amount of free area. Experiments have been made with such a device, and a description of it is being included in this report.

A New Method

It is apparent that there are numerous cases where it will always be hard to measure the air flow by the usual means, due to special conditions which make it difficult or impossible to apply the rules which have been discussed. The unit ventilator represents one case as already pointed out. In general, any case where the grille proper is preceded by any obstruction of such character that it is difficult to predict its effect on the air stream will fall into this class. Another case would be that of a unit heater of the suspended type using a disc or propeller fan. Finally, grilles of a highly ornamental pattern are difficult because of the difficulty in estimating the amount of free area and the average width of the frets.

Some effort has been made to develop a new device to be used in conjunction with the anemometer that will permit making reasonably accurate measurements without any knowledge of the details of the grille other than the gross area. The objective is to be able to make a traverse with the device and then apply a simple formula:

$$\text{cfm} = K'VA$$

in which

V = indicated velocity
A = gross area of grille
K' = a coefficient—preferably a constant but possibly varying with velocity

The first attempt was a simple cylinder about 10 in. long having the same diameter as the anemometer. The theory was that the air entering this cylinder from the grille would spread out and diffuse until by the time it reached the instrument on the far end of the tube it would have a uniform velocity across the whole area of the anemometer. The variation in K proved to be too great, being rather high with grilles of large free area, and small with those having small free area. The reason for this can be seen if it is realized that the cone and anemometer introduce a resistance so that when they are held up against any given spot on a grille, less air actually passes that spot than would be the case if the device were not there. Now consider two points equal to each other in area, one on a grille having 80 per cent free area, the other having 40 per cent. Imagine, further, that the same volume of air is being delivered at both points. It follows that since the second has only half as great a free area as the first, the actual velocity of the air will be twice as great, and the kinetic energy four times as great. Naturally, this second stream will be relatively unaffected by the presence of any resistance such as the anemometer and cone.

The problem then was of a somewhat paradoxical

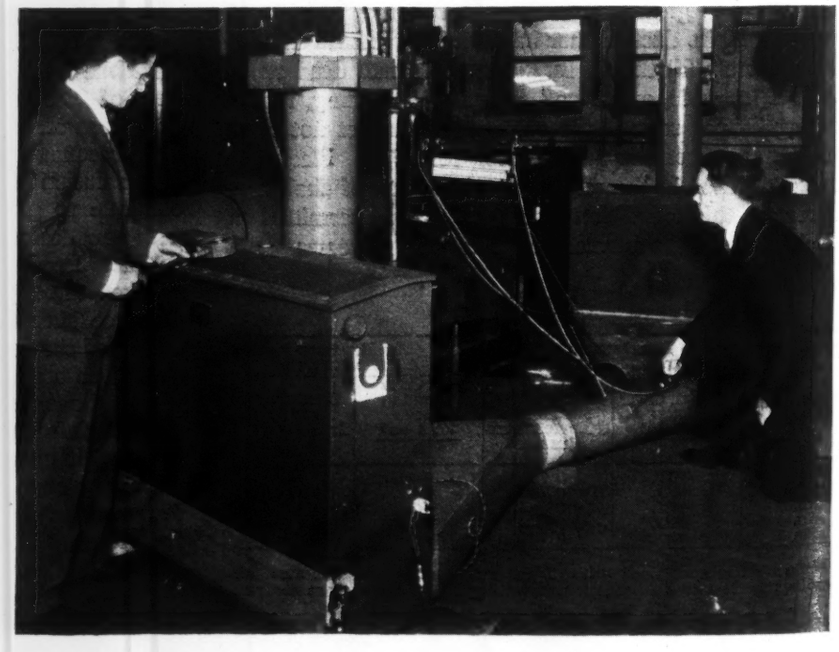


Fig. 10—Set-up for testing unit ventilators

nature. It was necessary to introduce resistance in order to break up and diffuse the jets coming from the interstices of the grille and at the same time keep the total resistance down to a minimum. After trying several types of cylinders, cones, and other special shapes, the one shown in Fig. 7 was finally evolved. At first only the right hand double conical portion was used, with the anemometer mounted on the outer end. The cone is fitted with a handle, not shown in the sketch. Later the 3 in. tube was added, made so that it simply slips over the outer end of the anemometer. In making a traverse the grille face is divided into strips approximately 5 in. wide and the cone held with its rectangular entrance in

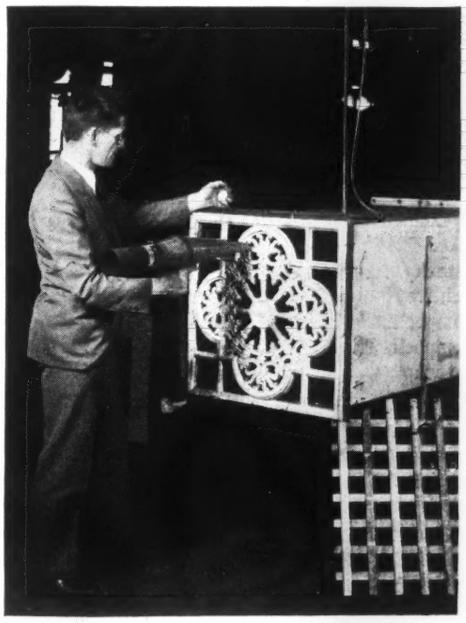


Fig. 11—Showing method of making traverse with anemometer mounted in special cone

contact with the grille and the axis at right angles to the face. The device is moved slowly from one end of the strip to the other.

It was decided that the long narrow rectangular opening would be most satisfactory in reaching the narrow dead strips along the grille edges that so frequently cause trouble because of the inability of a circular instrument to reach them effectively. However, it has been found that unlike the normal anemometer traverse, the instrument is quite sensitive to the variations in the angle of flow. It is believed that if the entrance is made more nearly square this effect will be greatly reduced, so this change is to be made in the next model. It will also be practical to have two or more different types of entrance cones, each adapted to a certain class of work. At first the entrance was made of such size as to give an average coefficient of 1.00. This was a mistake and when the entrance was reduced somewhat in size, considerably greater consistency was obtained. The screens are for the purpose of breaking up the individual air jets coming from the grille. Naturally, they introduce resistance, and it was for the purpose of partly counteracting this that the short 3 in. cylinder was added to the outlet side of the anemometer. The anemometer itself has somewhat over 20 per cent of its area occupied by the dial and supporting bars. Hence, adding a short length of 3 in. cylinder has the effect of increasing the outlet area. This, in turn, has the effect of giving a partial recovery of velocity pressure that is available for overcoming the screen resistance. The results obtained showed marked improvement when this cylinder was added. The complete device is, of course, somewhat large and slightly awkward to use, but it is thought now that it is possible to reduce the entrance cone both in diameter and length.

The instrument as illustrated has an average coefficient ranging from about 1.03 at 200 fpm to 1.15 at 800 fpm, with maximum errors of about 7 per cent. In judging the value of the instrument one must keep clearly in mind that these maximum errors occur under conditions where all other methods have failed completely to give even approximately correct results. For example, one test of particular interest was made with a suspended type unit heater. This was set against the end of a 24 in. duct in the position usually occupied by the grilles. The propeller fan was run so as to give the same velocity distribution that it would have in service, namely, a low velocity center, surrounded by a high velocity ring, and four relatively dead spots in the corners. When a traverse was made with this new device, picking the coefficient from a curve based on results obtained from several grilles covering a wide range of types, the calculated discharge from the heater was only 1 per cent in error.

Furthermore, there is every reason to believe that the accuracy of the device can be still further increased by further development. It is planned to continue these investigations and full details of the progress of the work will be furnished to the Society.

Exhaust Grilles

No additional data have been secured on exhaust grille measurements since the last report was made. The results secured at that time have been carefully studied, and a limited amount of similar data have been secured from other experimenters, as a result of which it has been concluded that the formula recommended at that time, namely

$$\text{cfm} = KVA$$

in the most accurate one possible at this time.

Most of the men applying this formula are using the average value 0.80 for K. With the exhaust grilles the air velocity had a much greater effect upon the value of the coefficient than was the case with supply grilles. For this reason, it is strongly recommended that the values of K given in THE A. S. H. V. E. GUIDE be used for the various velocities rather than the single constant.

No observations were taken with the lattice work grilles having large frets. However, the fact that the large plaster grilles gave results in very close agreement with the average figures obtained with the small lattice designs, indicates that on exhaust the large frets do not have any material effect. However, the usual precautions regarding the methods of making the traverse as already discussed under that heading, should be followed in order to get the best results.

CANDIDATES FOR MEMBERSHIP

The Constitution of the Society as now amended, requires the following mode of procedure in voting on applicants for membership in the Society. All applications for membership are to be sent to the Secretary and the names of applicants and their references shall be printed in the next issue of the JOURNAL of the Society or sent to the members in other approved manner as ordered by the Council. When replies are received from references, the Candidate's application shall be submitted to and acted upon by the Membership Committee as soon as possible.

When the Membership Committee has acted favorably upon a Candidate's application and assigned his grade, the Council shall vote upon the election of the proposed Candidate for membership by letter ballot. During the past month 14 applications for membership have been received and the names of these men and their sponsors are published in the following list.

Members are requested to scrutinize the list with care. The Membership Committee, and in turn the Council, urge the members to assume their share of the responsibility of receiving these candidates into membership by advising the Secretary promptly of any whose eligibility for membership is in any way questioned.

All correspondence in regard to such matters is strictly confidential, and is solely for the good of the Society, which it is the duty of every member to promote.

Unless objection is made by some members by September 15, 1933, these candidates will be balloted upon by the Council. Those elected to membership will be notified by the Secretary immediately after election.

CANDIDATES	REFERENCES	
	Proposers	*Seconders*
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	W. F. Uhl	J. B. Harris
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	W. W. Cox	W. E. Beggs
BOYLE, WALTER W., Consulting Engr., Pittsburgh, Pa.	L. B. Pittock	F. A. Gunther
	G. S. McEllroy	F. C. Houghten
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	E. K. Campbell, Jr.	Carl Clegg
COLLINS, JOHN F., JR., Supervisor of Steam Installations, Allegheny Co. Steam Htg. Co., Pittsburgh, Pa.	L. B. Pittock	G. S. McEllroy
	F. A. Gunther	F. C. Houghten
DEELY, JAMES J., Sales Engr., Brooklyn Union Gas Co., Brooklyn, N. Y.	J. M. Traugott	F. B. Herty (*A.S.M.E.*)
	J. M. Lednum (*A.S.M.E.*)	J. I. Quinn (*Non-Member*)
ERICSSON, ERIC B., Engr.-Custodian, Chicago Board of Education, Chicago, Ill.	John Howatt	R. M. Getschow
	J. J. Aeberly	J. J. Finan
GOODWIN, WALTER COOK, Div. Engr., Air Cond. Equip. Div., Supply Engrg. Dept., Westinghouse Elec. & Mfg. Co., East Pittsburgh, Pa.	Frank Thornton, Jr.	F. H. Hecht
	H. B. Orr	F. C. Houghten
HARTWEIN, CHARLES E., Htg. Engr., St. Louis County Gas Co., St. Louis, Mo.	K. S. Wallace	C. R. Davis
	C. A. Pickett	G. W. F. Myers
HERTY, FRANK B., House Htg. Supervisor, Bklyn. Union Gas Co., Brooklyn, N. Y.	Hugh Cuthrell (*Non-Member*)	C. G. Segeler (*Non-Member*)
	D. L. Caldwell (*Non-Member*)	F. M. Gahagan (*Non-Member*)
LYNN, JOHN HENRY, Bldg. Supt. & Chief Engr., Dallas Power & Light Co., Dallas, Tex.	T. M. Cunningham	O. E. Gammill, Jr.
	G. H. Meffert	F. C. Hersh
MILLER, LORIN G., Prof., Mech. Engrg., Michigan State College, East Lansing, Mich.	J. H. Van Alsburg	G. L. Larson
	S. H. Downs	H. B. Dirks (*A.S.M.E.*)
SWANSON, HARRY, Engr., The Fels Co., Portland, Maine.	W. T. Jones	A. B. Fels
	C. H. Mitchell	C. R. Swaney

Candidates Elected

In past issues of the JOURNAL of the Society the names of the following men were listed as Candidates for Membership. The membership grade of each Candidate has been assigned by the Membership Committee and balloted upon by the Council. We are now instructed by the Council to post herewith, as required by Art. II, Sec. VIII, of the By-Laws, the following list of candidates elected:

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NEU, H. J. E., Pres., Société Anonyme des Etablissements Neu, Lille, France.

SMITH, J. DARRELL, Mech. Engrg. Dept., Philadelphia & Reading Coal & Iron Co., Pottsville, Pa.

TOONDER, CLARENCE L., Air Cond. Engr., Kelvinator Corp., Detroit, Mich.

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SMYERS, EDWARD C., Sales Engr., Minneapolis-Honeywell Regulator Co., Pittsburgh, Pa.

JUNIORS

CORRAO, JOSEPH, Estimator & Engr., Herman Lawson Co., San Francisco, Calif.

SAWYER, J. NEAL, Field Engr., Holland Furnace Co., Kansas City, Mo.

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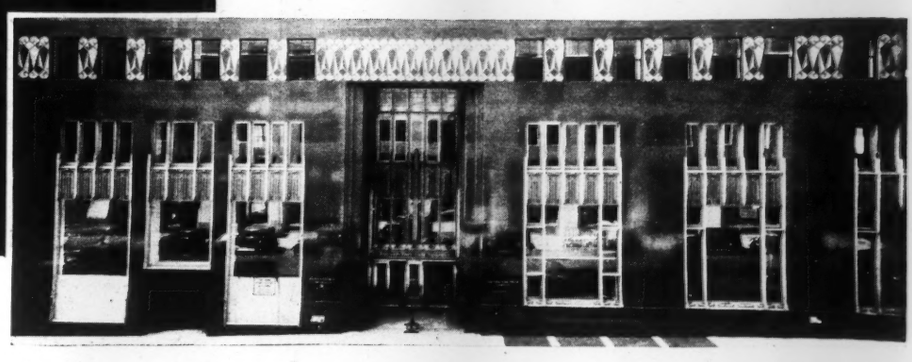

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Conditions Air—

(From page 465, Editorial Section)

tons. The equipment is so arranged that an additional 70-ton compressor may be installed later to bring the plant up to full capacity.

With cooling-tower water at 95 F, the condenser water requirements amount to approximately 6.75 gpm per ton and the steam consumption to approximately 35.75 lb per hour per ton. With city water not exceeding 70 F available for condensing purposes, the water requirement is approximately 2.85 gpm per ton and the steam consumption approximately 26 lb per hour per ton. The required steam pressure at the nozzles is 130 lb gage when using cooling-tower water and 100 lb gage when using city water. It is necessary, of course, that the steam be dry.

These tonnages and water and steam requirements are on the basis of cooling water to 52 F. Should the plant be operated at a lower evaporator leaving temperature, its tonnage capacity will, of course, drop off. For example, if the evaporator water temperature is 42 F, the capacity of the plant is 115 tons as compared with 180 tons with evaporator water at 52 F. The total steam and condenser water requirements operating at 42-F evaporator water for 115 tons refrigerating capacity are the same as when operating at 52-F evaporator water and 180 tons capacity.

The cooling tower is designed to cool 960 gpm of water from 110 F to 95 F when the outside wet bulb does not exceed 75 F. This amount of condenser water is sufficient to permit the refrigeration equipment to operate up to 140 tons cooling capacity, (52-F evaporator). In selecting this capacity as the most economical for the present installation, the duration of peak loads was weighed against tower fan and pump horsepower requirements.

The amount of heat per ton to be removed from a steam-jet refrigeration condenser exceeds by approximately three times the heat per ton to be removed from a compression refrigeration condenser. The burden on the cooling tower is not in the same proportion, however, for water in the steam installation may be run to higher temperatures with resulting higher efficiency of heat transfer in the cooling tower.

City Water Aids Cooling Tower

When the load exceeds 140 tons, city water is fed into the cooling-tower pump suction in amounts as necessary to help the cooling tower over the peaks. This added city water would ordinarily be wasted through an overflow, but the cooling-tower washer tank is below the sewer level. In order to avoid a costly sump-pump arrangement, it is necessary to waste the excess under pressure. This is done from the water line from the condenser to the washer sprays through an automatic electric valve which is controlled from a float switch operating on the rise and fall of water level in the washer tank. The action is graduated; that is, the valves will assume positions between full open and full closed, as required by the amount of water to be wasted.

Water hammer and surging in the sprays are thus avoided.

Discharge of Air from Cooling Tower

The air leaving the cooling tower is at temperatures ranging up to approximately 95 F saturated. When this air reaches the shaft some of the moisture it contains condenses to form a "steam". However the "steam" is not dense or particularly objectionable. Moisture does form on the walls of the shaft and in the escape passageways through the shaft at each floor, but not in amounts which cannot be dried out by a short run of the cooling-tower fan after the sprays have been shut down at the end of the day.

Balancing Inside-Outside Conditions

The inside air conditions are guaranteed not to exceed 80-F D. B. and 50 per cent relative humidity when outside conditions do not exceed 95-F D. B. and 75-F wet bulb. To maintain these conditions on the eight floors treated by the system, not more than 160 tons of the 180 tons refrigeration effect is required and the evaporator water temperature need be no lower than 52 F. But as outside conditions drop, it is desirable that lower inside conditions be maintained until at 70-F D. B., 70-F W. B. outside, the inside conditions should be at 70-F D. B. and 50 per cent relative humidity. This lower inside condition represents a lower maintained dew-point and therefore makes necessary operating the refrigeration equipment at a lower evaporator temperature; 42 F in this case.

As pointed out above, at this evaporator temperature, the capacity of the refrigeration plant has dropped to 115 tons. Until the outside wet-bulb temperature falls below 70 F, there is only the decrease in transmission heat load to offset this drop in available tonnage. As the outside wet bulb falls below 70 F, the heat load represented by fresh air decreases, and a further advantage is gained.

The transmission load on this job amounts to about 30 per cent of the total load or about 48 tons. The balance, consisting of the internal heat load and fresh air load (the fresh air load changes but little between 75-F W. B. and 70-F W. B. as the inside maintained wet bulb decreases) can be cared for by the 115 tons cooling effect available at the lower evaporator temperature. Of course, lag in the transmission load makes it impossible under ordinary conditions to synchronize outside and inside conditions closely. Only prolonged, approximately constant, outside conditions would make this possible.

Space Required by Conditioning Plant

Because floor area in a modern office building is so valuable, the amount of space required by the equipment is of great importance. On this job the refrigeration and cooling tower equipment in the basement occupy approximately 900 sq ft of space with 13-ft headroom. The fans and dehumidifier equipment on the office floors occupy a total of approximately 1775 sq ft. The total area conditioned is approximately 64,000 sq ft.

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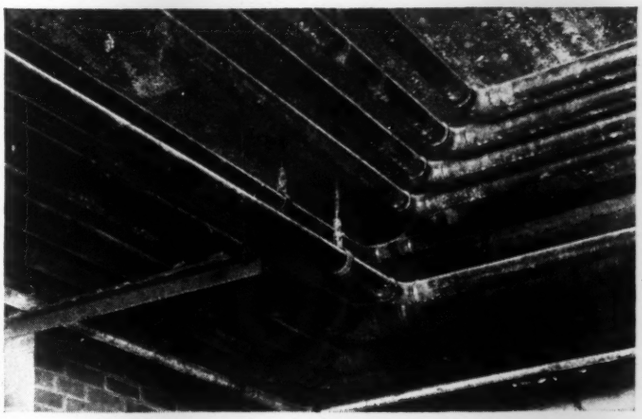

● Taylor Forge Wrought Iron Weldells make all-welded wrought iron lines possible. These correctly engineered fittings are made from Byers Genuine Wrought Iron Pipe, not plate, by the patented Taylor Forge process.

Check these advantages and remember that you pay no price premium for them. Every Taylor Forge Wrought Iron Weldell has a smooth interior and exterior. Ends are machine beveled to make a neat job. Tangents assure proper lining up. Selective reinforcement for greater strength. A request will bring complete information and estimating prices.

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That's the gist of the letters we get from subscribers to HEATING, PIPING AND AIR CONDITIONING. They're from heating, piping and air conditioning engineers all over the country, working in all types of construction.

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HEATING, PIPING AND AIR CONDITIONING
1900 Prairie Avenue, Chicago, Illinois

Unit Heaters—

(From page 449, Editorial Section)

centers, to give a rugged construction. This construction with ground surface joints and a high-temperature superheat gasket permits working pressures up to 125-lb, the $\frac{3}{16}$-in. copper headsheet with a special brazing process

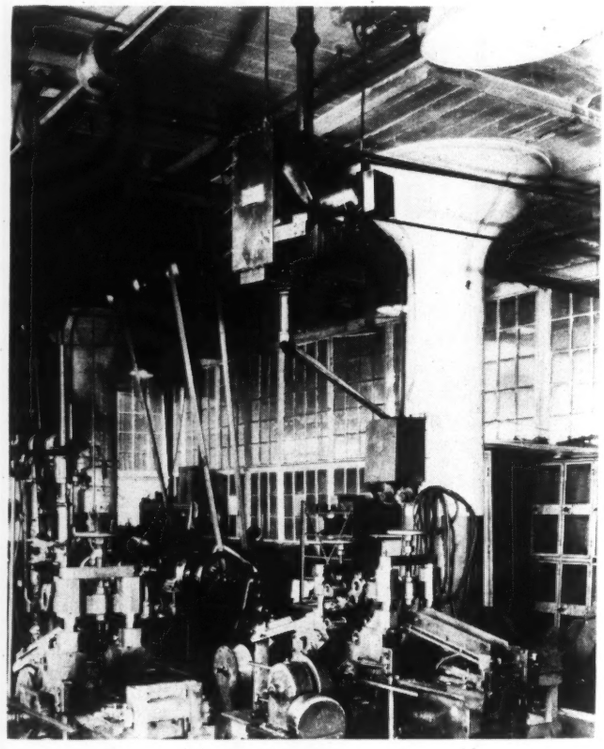

Fig. 4—Unit heaters installed in a large plant

providing a tube connection which resists large strains. Particular attention has been paid the reduction of noise and vibration.

Text on Fundamentals of Heat

To some extent, the title "Elements of Industrial Heat" is a misnomer for the new text-book by John A. Randall, President of the Rochester Athenaeum and Mechanics Institute and J. Warren Gillon, Instructor in Heat and Mechanics at the Institute. (261 pp., $2.75. John Wiley & Sons, Inc., New York City.) While it is an excellent and up-to-date text on the fundamental principles of heat, no particular attention is devoted to industrial applications. The material included is intended to develop a good preparatory knowledge of heat engineering to serve as a background for further study.

The chapter headings include fundamental concepts; calorimetry (of solid, liquid and gaseous fuels); expansion of solids, liquids and gases; the three states of matter; steam calorimeters; conduction, convection, and radiation; fuels and their combustion; properties of air and its moisture content; and energy diagrams. The chapter on air briefly discusses humidification and explains the Carrier psychrometric chart. A feature of the book are the many problems presented in the text and in two appendices, planned to aid in understanding the text and in reviewing it.

AEROFIN

IS THE HEAT-SURFACE FOUND IN THE

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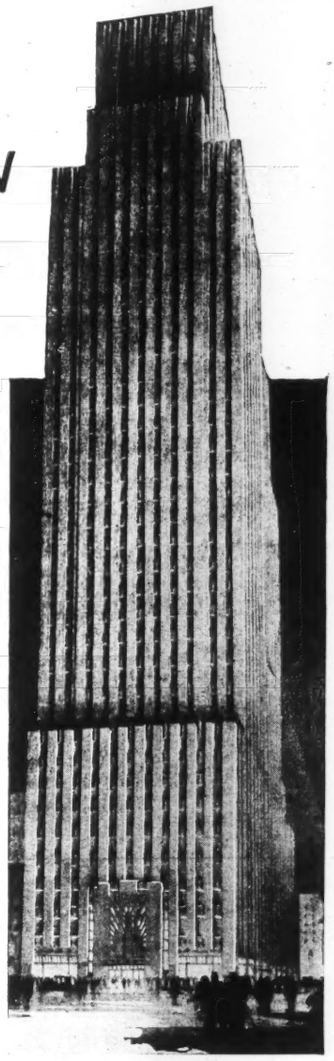
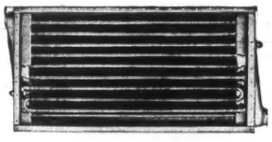
NEW YORK CITY





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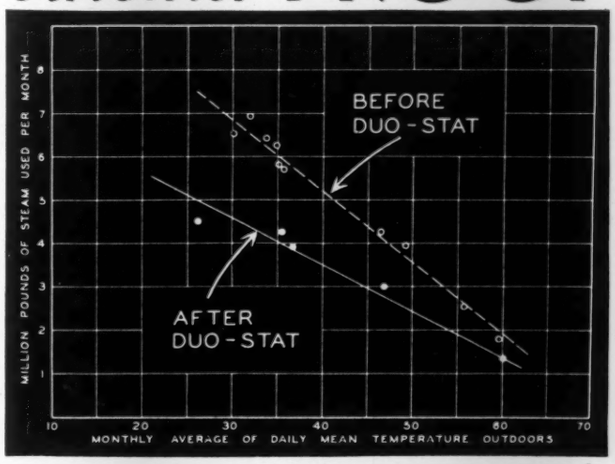
Another PROOF


MILLION POUNDS OF STEAM USED PER MONTH
BEFORE DUO-STAT
AFTER DUO-STAT
MONTHLY AVERAGE OF DAILY MEAN TEMPERATURE OUTDOORS
10 20 30 40 50 60 70

—*an accurate picture of* DUO-STAT SAVINGS *in the* HEARST BUILDING of Chicago

IT was even more than we had hoped for! In exactly six weeks the Raymond Duo-Stat in the Hearst Building, Chicago, had *paid* for itself. The story of this big cut in steam costs is shown above in the simplest possible way—by simply plotting steam used per month against the daily mean temperature outdoors.

If this were the only record of its kind you would have a right to question it. But the record of the Hearst Building is merely one leaf of a great volume of evidence for the Raymond Duo-Stat. We have charted many of these records in the same way that the Duo-Stat record in the Hearst Building is pictured above. Have you received a set of these charts?

What would the picture of Duo-Stat savings be in the building, or buildings, in which you are interested? We will gladly chart this for you if you will simply supply a record of steam or fuel consumption (by months) for the last year. We have prepared many charts of this kind, and the theoretical steam saving that they show has invariably been exceeded by actual performance in those cases where the Duo-Stat was later installed.

F. I. RAYMOND CO.
629 W. Washington Blvd.
CHICAGO, ILL.

The DUO-STAT *alone combines these features*

- Controls room temperature by a combination of outside and radiator temperature. (A fully protected principle.)
- Is just as effective with one-pipe system as two-pipe system.
- Accomplishes partial filling of radiators in either a one-pipe or two-pipe system.
- Cannot be upset by open windows.
- Gives a modulated control that can only be described as Duo-Stat modulation.

RAYMOND DUO-STAT

EASY TO GET AT!

ALL PARTS ACCESSIBLE

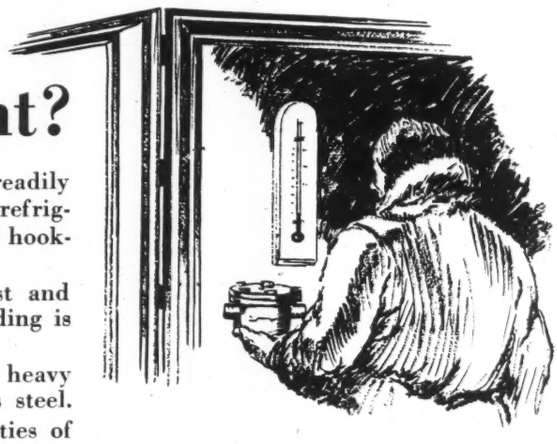
When most unit heaters are installed, sufficient space must be left at one end of the heater for withdrawing the heating coil. But this space is not necessary with Clarage Unitherms. They can be installed in very close quarters since, as shown at left, the coil is removable through the front of a Clarage Unit—short, simple job. Also note that the fan housings can be removed. Thus every operating part is accessible—easy to get at.

ECONOMY in first cost and in operation, comfortable working conditions and years of dependable service follow the selection of Clarage Unitherms. They give you uniform heating over extremely wide areas — without cold spots or hot blasts. They are flexible — heat where you want it and when you want it. They are compact—greater B.t.u. capacity. They have centrifugal fans designed with a self-limiting horsepower characteristic. Built in sizes to cover small or large requirements at steam pressures up to 200 pounds. We welcome your inquiry. CLARAGE FAN COMPANY, Kalamazoo, Mich.

CLARAGE

UNITHERM UNIT HEATERS

What? A Steam Trap in a Refrigerating Plant?



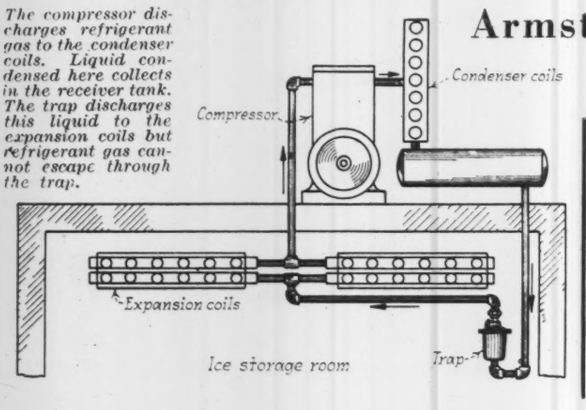
"BELIEVE it or not," Armstrong Steam Traps can be readily adapted for refrigerant service when all the liquid refrigerant can be carried on the low or evaporating side. A typical hook-up is shown below.

In such installations, the two big advantages are low cost and reliable service. No closed float to leak or collapse, gas binding is eliminated, and valve parts do not wire draw.

For ammonia service, Armstrong trap bodies are made extra heavy and all interior parts, including the bucket, are of stainless steel.

Armstrong representatives will gladly discuss the possibilities of Armstrong Traps for refrigerant service. Get in touch with your local representative or write us direct.

The compressor discharges refrigerant gas to the condenser coils. Liquid condensed here collects in the receiver tank. The trap discharges this liquid to the expansion coils but refrigerant gas cannot escape through the trap.



Armstrong Machine Works
Three Rivers, Mich.

ARMSTRONG "INVERTED BUCKET" STEAM TRAP

Liquid Level Control

IN many cases it is not practical to carry all the liquid refrigerant on the low pressure or evaporating side. For such service Armstrong offers an ammonia liquid level control which admits liquid ammonia (or other refrigerant) to the evaporating coil as fast as it is required and maintains a pre-determined liquid level in the evaporating coil. Blueprints gladly furnished on request.

Specifications for Traffic Tunnel Fans

IN the August issue of HEATING, PIPING AND AIR CONDITIONING, in the article by J. E. Gill on testing the fans for the East Boston traffic tunnel, a statement is made as follows: "The specifications incorporated a provision not heretofore used on jobs of this nature, etc."

In July, 1925, the specifications were written for the Holland Tunnel fans. On page 5 of these specifications will be found the following:

"Prior to the award of the contract, tests of one fan, similar to each type submitted by the bidder, except transformer blower fans, will be required, etc.

"If the above tests show the brake horsepower to exceed by 3% or more in the operating range, or by 5% at any other point, the brake horsepower shown on the performance curves submitted, the bids may be rejected. Operating range of fan is from 30% to 85% of wide open volume."

It will be noted from the above "a provision not heretofore used on jobs of this nature" was used eight years previously in the Holland Tunnel specifications.—*A. C. Davis.*

Author's Reply

The Holland Tunnel specifications called for a test of a fan similar to each type submitted by the bidder prior to the award of the contract and with the purpose of verifying the curves submitted with the bids. This clause was similar to that incorporated in the Boston Tunnel specifications but differed in these important features:

Whereas, the Holland Tunnel specifications stated:

"The tests are to be made *by the manufacturer at his plant under the supervision of the Engineer* and in accordance with the standard test code of the National Society of Heating & Ventilating Engineers adopted June, 1923, subject to such modifications as have been adopted since then," the Boston Tunnel specifications stated—

"The test will be made *in a laboratory, in the vicinity of the City of Boston,* to be designated by the Department," and "The tests on this fan are to be made *under the supervision of the Engineer in the presence of the manufacturer.*"

Furthermore, the following clauses were contained in the Boston Tunnel specifications which were not present in the Holland Tunnel specifications:

"2. In addition to the above tests at least two points in the performance range shall be tested by means of a flow nozzle," and "should the capacity of the fan as shown by the flow nozzle differ by more than 3 per cent from the capacity as shown by the impact tube under similar conditions, the bid submitted may be rejected."

It was these important differences and provisions "not heretofore used on jobs of this nature" to which reference was made. —*J. E. Gill.*

Conventions and Expositions

American Gas Association: Annual convention, September 25-29, Chicago. Headquarters Office, 420 Lexington Ave., New York City.

International Acetylene Association: Annual convention, September 26-29, Congress Hotel, Chicago. ("International Acetylene Day" at *A Century of Progress* to be held during meeting.) Headquarters Office, 35 E. 42nd St., New York City.

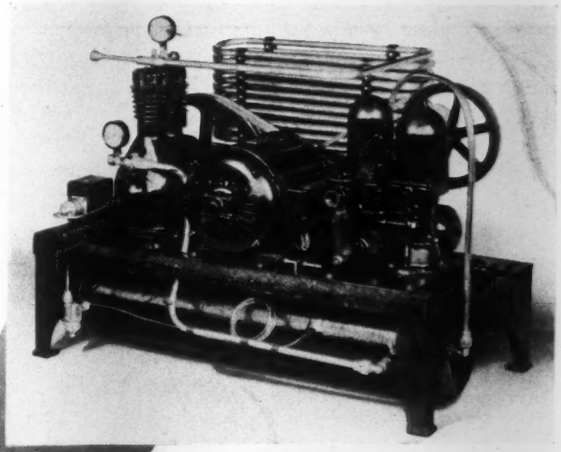
American Welding Society: Fall meeting, October 2-6, Book-Cadillac Hotel, Detroit, Mich. Headquarters Office, 29 W. 39th St., New York City.

National Association of Practical Refrigerating Engineers: Annual convention and educational exhibition, November 8-11, Roosevelt Hotel, New Orleans, La. National Secretary, Edward H. Fox, 435 N. Waller Ave., Chicago.

American Society of Heating and Ventilating Engineers: Annual meeting, February 5-8, New York City. Secretary, A. V. Hutchinson, 51 Madison Ave., New York City.

International Heating and Ventilating Exposition: Third exposition, February 5-9, 1933, Grand Central Palace, New York City. Charles F. Roth, Manager, International Exposition Co., Grand Central Palace, New York City.

WAGNER
Motors for Air Conditioning



Repulsion-Induction
Split-Phase
Capacitor
Direct-Current
Squirrel-Cage
Fynn-Weichsel Synchronous

THE motor operating the cooling unit of an air conditioning system illustrated above, is a Wagner repulsion-start-induction.

Other types of motors Wagner builds for air conditioning machinery are: split-phase (primarily for small blowers and pumps), capacitor, squirrel-cage (3-phase and 2-phase), direct-current—and, on large jobs, the powerfactor-correcting Fynn-Weichsel synchronous motor.

Whatever your motor-driven product may be, there's a type of Wagner motor available to drive it.

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Please send literature on the types of Wagner motors checked below.
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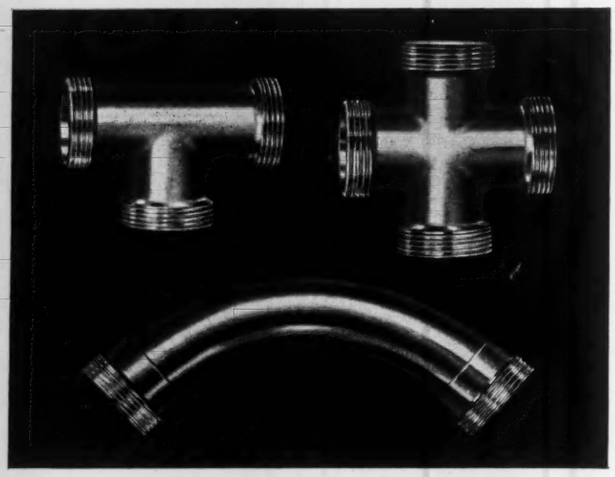

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Equipment—

(From page 472, Editorial Section)

protects the moving parts. The condenser is of the conventional copper-fin type design.

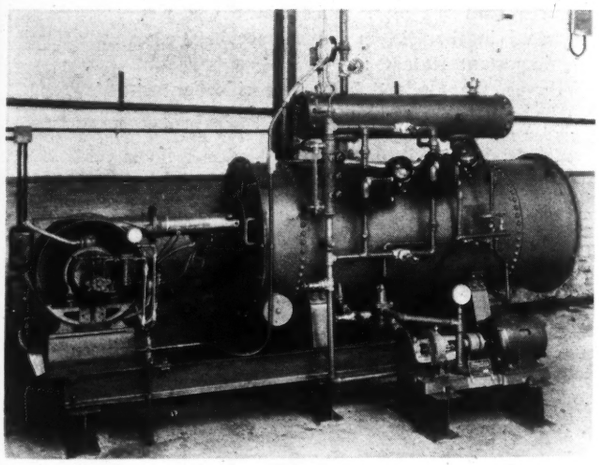
Operation is controlled by means of a suction pressure switch mounted directly on the unit, this incorporating an overload safety device. The large-capacity, cylindrical-type liquid receiver is mounted on the base. It is equipped with gas and liquid stop valves and fusible safety element.

A magnetic relay switch is mounted directly on the unit and completely wired to the motor and controls. The suction gas strainer is mounted directly on the compressor. The unit runs at slow speeds.

The new condenser is for use in conjunction with the new freon compressors announced a few months ago.

Compact Oil-Burning Boiler

Cleaver-Brooks Company, Milwaukee, Wis., has introduced several sizes of oil-burning steam boilers designed to operate with little labor, in small space and at low cost. The type of construction used eliminates the need of a foundation. The

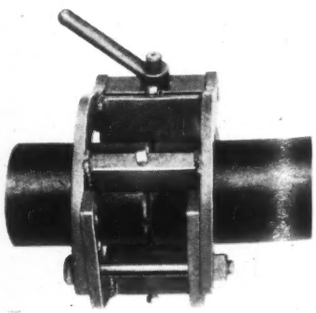


boiler and burner are an integral unit each designed to operate with the other to promote high overall operating efficiencies.

All air delivered into the combustion chamber is handled by a blower to give a controlled predetermined mixture of air and oil aimed to insure perfect combustion and complete use of fuel. Operated with automatic controls, the burner runs intermittently to take care of fluctuating loads and generates only the amount of steam required. Sizes range from 5 to 500 boiler hp in working pressures from vacuum to 200 lb. They may be used for building heating as well as in the plant by the installation of pressure-reducing valves.

Offers Pipe Welding Clamp

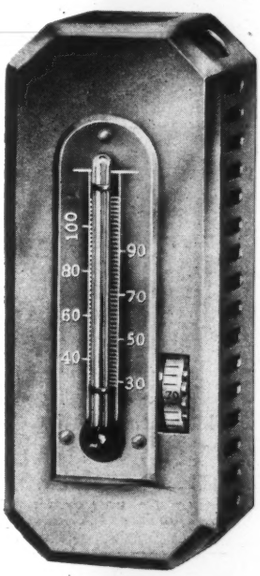
A new pipe welding clamp has recently been announced by Oster-Williams, 2057 E. 61st Place, Cleveland, Ohio.



The clamp is adjustable for all sizes within its range and is simple in design. It is made at present in two ranges, one from 4 to 6 and the other from 8 to 12 in. Other sizes will be available in the future. One lever controls the action of the clamp, making it easy to put on and take

"GENUINE DETROIT"

Complete AIR CONDITIONING CONTROLS



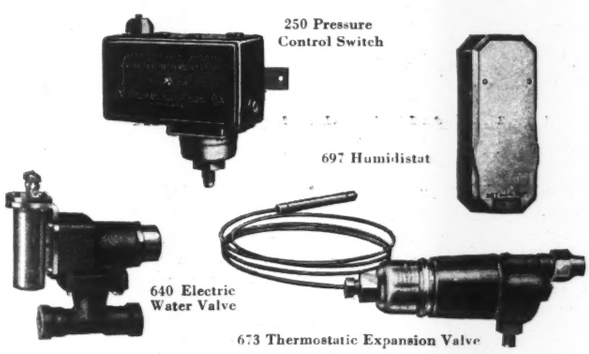
No. 444 Thermostat

Low-priced dependable controls

WHETHER your installation calls for complete air conditioning or automatic control of only heating and humidity, there are "Genuine Detroit" accurate, reliable, low-priced controls ready for any purpose. Before you plan the job, find out how little it will cost to include these tested controls.

The Complete line of "Genuine Detroit" Air Conditioning Controls includes with others the No. 444 Thermostat, No. 697 Humidistat, No. 673 Thermostatic Expansion Valve, No. 640 Electric Water Valve and the No. 250 Control Switch.

Write for Bulletins Nos. 39-B, 51, 50 and 61.



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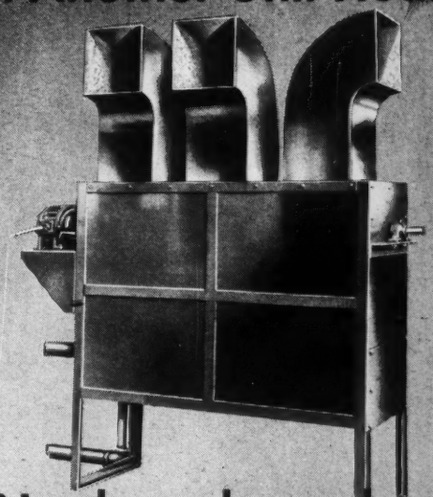
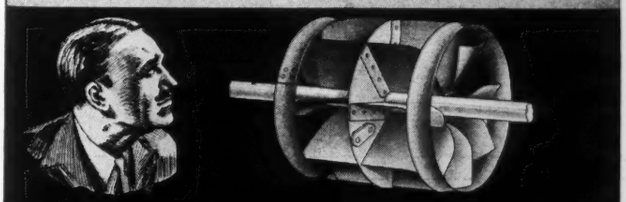


Not by a long shot!



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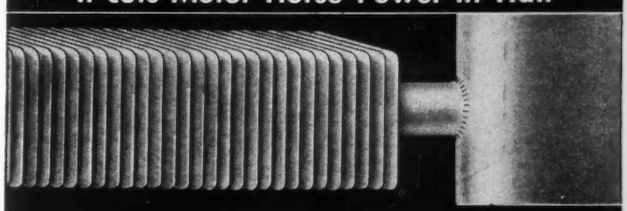
It cuts Motor Horse Power in Half



And this NEW LEAK-PROOF Heater
Just try to break one of the joints!

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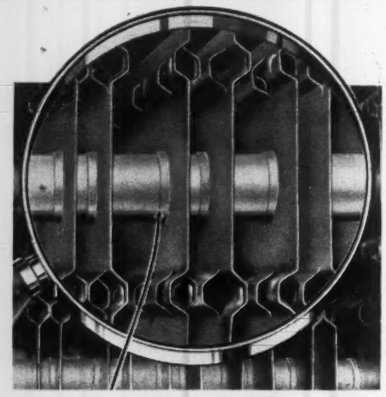

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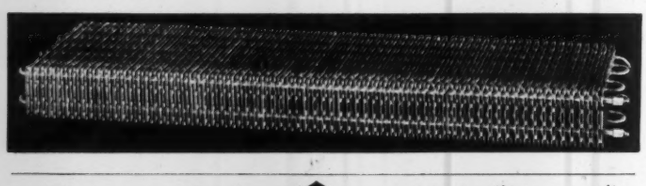


Always marked with the "Diamond"

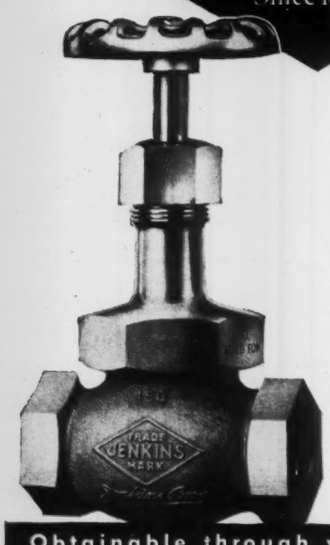
Jenkins

BRONZE IRON STEEL

VALVES

Since 1864

NRA MEMBER U.S. WE DO OUR PART



THERE is QUALITY in Fig. 106-A Jenkins Bronze Globe Valve—extra deep stuffing box—slip-on, stay-on disc holder—one-piece screw-over bonnet—Jenkins disc. It pays to insist on Jenkins Valves and the extra service that has characterized Jenkins quality since 1864.

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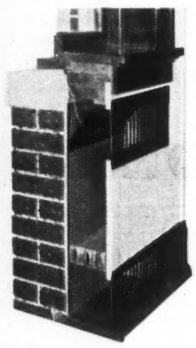
Fig. 106-A Globe, Screwed

Obtainable through your supply house

POSITIVE *Air Circulation*

The Most Practical Scientific Method



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Young "Streamaire" copper core with elliptical tubes and integral fins is much copied but never duplicated in its sturdier construction and heat transfer qualities.

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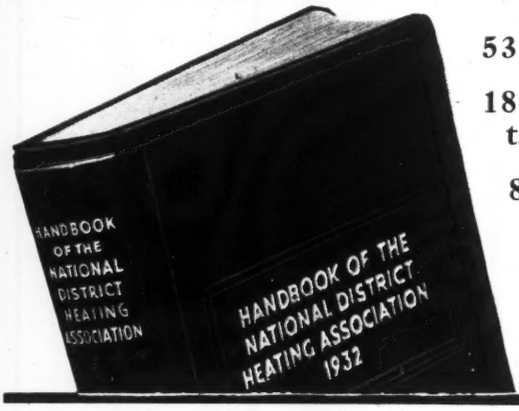

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THE HANDBOOK of the National District Heating Association will save you time and money in the design, operation and maintenance of central and district heating systems. The sole purpose of this authority is to provide a complete manual of practice for the use of the chief and operating engineers, for consulting engineers, architects and building managers, and for all others engaged in district central heating work. Over 50 engineers—each a specialist in his own line—have collaborated to make it an outstanding volume.

The Handbook contains 50 pp. on steam meters and metering; 33 pp. on the economical use of steam; 26 pp. on corrosion of steam heating systems; 93 pp. on steam distribution; 28 pp. on building heating methods; 65 pp. on district heating plants; and over 200 more pages on rates, sales activities, heat transfer, determining annual steam or fuel consumption, etc.

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HEATING, PIPING and AIR CONDITIONING

1900 Prairie Ave. Chicago, Illinois

off the pipe. It automatically centers the two lengths of pipe to be welded and adjustable screw spacers keep them the correct distance apart for efficient welding. The pipe is held rigidly. Open construction of the clamp leaves large space for tacking. Welded steel construction is used to give maximum strength with minimum weight. The smaller tool weighs 18 lb and the larger 26 lb.

Inverted-Bucket Steam Trap

A new inverted-bucket trap particularly applicable to services where a large amount of condensate must be handled is now offered by Jas. P. Marsh Corporation, 2073 Southport Ave., Chicago. The trap is self venting and has a large water capacity. The motive element is an inverted seamless open bucket operating through a cast brass lever arm to open and close the swivel pin valve. Operation and construction are simple.

Body and cover are of cast iron. Bucket is of seamless copper and levers and clevises are of cast bronze. Pins, valve and seat are stainless steel. Design incorporates a high lever ratio for large capacities. All piping connections are made to the body which permits removal of the trap cover without disturbing piping. The entire operating mechanism lifts out with the trap cover making all parts accessible.

Standard finish is gloss black. ½ and ¾-in. traps can be furnished in satin finish or polished finish, nickel or chrome plate on special orders. This trap is suitable for working pressure up to 125 lb per sq in. and is made in three sizes—½, ¾ and 1 in.

Recent Trade Literature

Air Conditioning: American Radiator & Standard Sanitary Corporation, 40 West 40th St., New York City. 30 page booklet summarizing the reasons for air conditioning for comfort in an elementary manner.

Air Conditioning: Campbell Metal Window Corporation, 100 E. 42nd St., New York City. 20-page booklet dramatizing the importance of air conditioning and noise prevention and describing conditioning units which circulate and introduce fresh air, filter, and control noise in summer and winter; cool and dehumidify in summer; and heat and humidify in winter.

Air Conditioning: Frigidaire Corporation, Dayton, Ohio. 32-page booklet on relief from hay fever and other disorders by means of air conditioning, describing how air conditioning relieves hay fever and pollen asthma, summarizing medical research, and describing the operation of a Frigidaire air-conditioning system and how it affects bodily comfort.

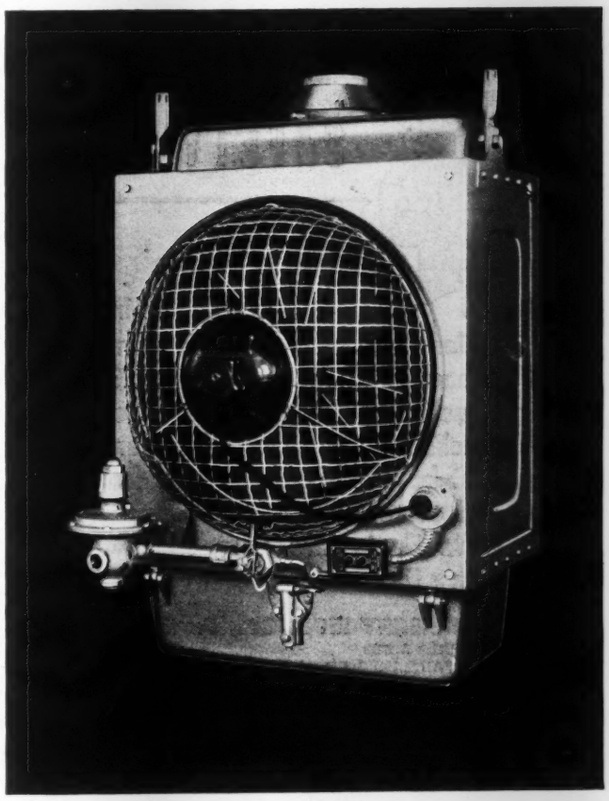
Compressors: Worthington Pump and Machinery Corporation, Harrison, N. J. 4-page bulletin illustrating the construction features of compressors for refrigeration (vertical duplex, horizontal single and duplex, steam-driven and motor-driven, diesel and gas-engine driven, ammonia, freon and CO_2 units, vacuum—steam-jet—equipment).

Fans: Buffalo Forge Company, Buffalo, N. Y. 4-page bulletin describing baby conoidal ventilating fans, giving dimensions, and rating tables. Several uses (ventilating, exhausting, blowing, cooling, drying) are pictured.

Furnace Fans: Air Controls, Inc., 1962 W. 114th St., Cleveland, Ohio. Selector chart showing cfm requirements of fan against furnace capacity, firepot size, gravity pipe area rating, or Btu per hr with a column for selecting the proper Miles automatic furnace fan or blower. Separate columns are provided for industrial and domestic service.

Heating Equipment: Jas. P. Marsh Corporation, 2073 Southport Ave., Chicago. Loose-leaf catalog of industrial instruments and heating equipment, giving complete information on packless radiator valves, thermostatic radiator traps, unit float and thermostatic traps, equalizer and boiler return trap, steam traps and vent, strainers, radiator valves, gages and thermometers, cocks, automatic air valves, air and vacuum valves, etc.

Gas Heating



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Economical in first cost—economical in operating cost—and economical in maintenance cost—Buffalo Gas Unit Heaters provide CLEAN heat in hundreds of industrial plants in every branch of industry.

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BY specifying Dart Unions you eliminate the element of chance from your pipe line installations. Two bronze seats accurately machined and carefully ground protect both sides of the joint from rust and corrosion. Take no chances . . . write Darts into your specifications every time.

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Humidity Instruments: Amthor Testing Instrument Co., Inc., 309 Johnson St., Brooklyn, N. Y. Data sheet describing direct-reading precision hygrometers, recorders, and other instruments for determination of air conditions.

Motors: General Electric Company, Schenectady, N. Y. 4-page data sheet on fractional-horsepower, single-phase, repulsion-start induction motors showing features, details of construction, etc.

Motors: Wagner Electric Corporation, 6370 Plymouth Ave., St. Louis, Mo. Bulletin sheet on vertical motors, illustrating and describing three different types and explaining by means of illustrations and applications how horizontal-type motors can be mounted vertically to save cost of special construction.

Motors: Westinghouse Electric & Manufacturing Company, East Pittsburgh, Pa. Leaflet on resistance split-phase motors, recommended for such applications as oil burners, fans and blowers, centrifugal pumps, etc.; applications, features, and construction are described. 4-page bulletin on high-speed synchronous motors with thermoguard protection for constant-speed machinery such as fans, blowers, compressors, pumps, etc. 4-page bulletin on splash- and drip-proof drives (motors and control), explaining construction in detail.

Nickel-Clad Steel: H. Boker & Co., Inc., 101 Duane St., New York City. 12-page technical bulletin on methods for the fabrication of nickel-clad steel sheets, strips, and wire.

Piping: Benjamin F. Shaw Company, Wilmington, Del. 32-page book illustrating this company's facilities for the fabrication and installation of piping in industrial and power plants and picturing numerous representative jobs.

Pumps: The Nash Engineering Co., South Norwalk, Conn. 36-page catalog of pumps:—air compressors and vacuum pumps, vacuum return-line heating pumps, condensation pumps, centrifugal pumps, suction sump and sewage pumps, and sewage ejectors. Descriptions, capacity tables, selection tables, etc., are included.

Pumps: Morris Machine Works, Baldwinsville, N. Y. 16-page bulletin on side-suction volute pumps, describing construction in detail, illustrating various types of drive, and giving rating tables and performance curves. Sizes are 1 to 20 in., capacities up to 14,000 gpm at heads to 120 ft. Suitable for handling clear water and abrasive or pulpy mixtures, with applications in industrial plants, chemical plants, pulp and paper mills, general pumping or unwatering services.

Pumps: Worthington Pump and Machinery Corporation, Harrison, N. J. Bulletins on horizontal single piston pattern steam pumps and horizontal simplex direct-acting piston pumps, giving complete information, capacity data, etc.

Solenoid Valves: Automatic Switch Co., 154 Grand St., New York City. Loose-leaf bulletin on solenoid valves for air, gas, steam, and liquids for automatic and remote control. Service recommendations, connections, specifications, and descriptions of several types are given.

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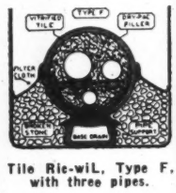
is waterproofed, thus keeping steam pipes dry and maintaining the highest thermal efficiency. Write for complete details.



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Valve Trim: The Edward Valve & Manufacturing Co., East Chicago, Ind. 12-page pamphlet describing the suitability and use of "Evalloy" as a valve trim material for high-temperature service; it is a stainless iron or 11-14 per cent chrome steel.

Venting Valves: Hoffman Specialty Company, Waterbury, Conn. New price list dated July 31. Also series of four bulletins giving descriptions, engineering data, and illustrations of complete line of heating specialties.

Water Treatment: D. W. Haering & Co., Inc., 3408 W. Monroe St., Chicago. 16-page booklet on water treating describing the applications of H-O-H treatments to various water problems and explaining causes and effects of problems common to boiler and hot-water equipment, air-conditioning equipment, refrigeration equipment, etc.

Welders: Universal Power Corporation, 1719 Clarkstone Rd., Cleveland, Ohio. 4-page bulletin on shunt inductor welder motor-generator sets. Also 8 page bulletin on welding accessories and clothing.

Welding Fittings: Grinnell Company, 277 W. Exchange St., Providence, R. I. 24-page catalog of welding fittings, including dimensional drawings, dimensions, illustrations and list prices. Among the fittings are elbows and returns; tees and crosses; reducers; caps, sleeves, and saddles; welding outlets; threaded outlets; forged steel flanges; lap welding necks; lap flanged welding cups; welding neck flanges; slip-on flanges; miscellaneous welding items, and pipe bends for welding.

Welding Fittings: Midwest Piping & Supply Company, Inc., St. Louis, Mo. 4-page leaflet describing welding fittings made from Reading puddled iron, including van stone welding nipples, 90-deg and 45-deg ells, welding heads, shaped welding nipples, and reducers.

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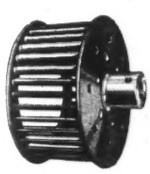

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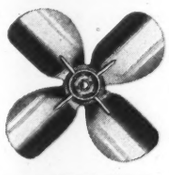

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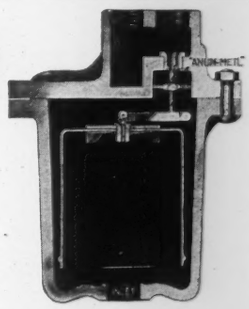

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